ROTARY VALVE ENGINES

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by

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ILLUSTRATED BY NUMEROUS DIAGRAMS
TABLES AND PHOTOGRAPHS

JOHN WILEY & SONS, INC.
NEW YORK

PREFACE

It is the author's experience that there is a wide demand for information on the various types of rotary valves which have from time to time been proposed, experimented with and produced during the past sixty years.

This book has been prepared for the purpose of embodying in one cover this information, together with the elementary fundamental principles it is necessary to employ for a successful design.

In a study of the practical examples it is hoped that the reader will find the treatment of some value to further progress and an inspiration to both student and practising engineer so that the remaining difficulties associated with the various forms of rotary valves may be overcome.

Appreciation is expressed to editors, societies, individuals and firms who have given valuable aid by the contribution of data and illustrations to make this treatise possible.

M.C.I.H.

September 1946

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INTRODUCTION

A VALVE by definition is a device for opening and closing a passageway, and a rotary valve is one that operates with a rotary or semi-rotary motion. It may not be general knowledge that such a valve can be used instead of a poppet-valve on a motor-car engine, and in place of the slide valve on a steam engine. During the evolution in the design of the internal combustion engine, the steam engine and the automobile, the incorporation of a rotary valve has at intervals been proclaimed as an improved feature of construction. Until recently, with few exceptions, after a comparatively short period the feature has been withdrawn from individual production, or the experimental work has been suspended. At each epoch the improved variation has been heralded with triumph and the same virtues of silence, freedom from vibration and elimination of inertia troubles have been extolled. The advantages of quiet operation and the abolition of reciprocating valve parts, subject to inertia stresses with the accompanying vibration, are beyond dispute.

The question, however, may be asked: "What were the real causes for the cessation of further experimental work?" and "What were the actual weaknesses in design?" An attempt will be made in the following pages to provide answers to the questions, and to outline the present stage of development.

It is only natural, but unfortunate, that the publicity for any new invention seeks to boost the desirable features but, at the same time, attempts to camouflage details which may be of doubtful value, and the data which could be of use to others are invariably hidden away in the back room and seldom see the light of day. If only part of this information is made available, progress in the art will be more rapid.

Experience tells us that many types of error in design, and also the need for detail improvements, can only be uncovered in production and sometimes only after prolonged service or testing. On the other hand, a superficial examination of past designs of rotary valves discloses fundamental deficiencies which are easily recognizable from a bare inspection of the meagre records available, and the existence of these faults is readily confirmed by simple calculations. Due prominence will be given to such points, and other departures from sound engineering, not quite so obvious, will be brought to light.

It is not to be expected that designers and manufacturers can reach fair conclusions on the many problems associated with rotary valves without having at their disposal information about the

relative merits of early designs. This information may exist at present in the scattered form available only to the few. Brought together, it can be of assistance in guiding design policy along the most progressive lines. The desire for reticence on the part of inventors and producers is not justified in any way. The ventilation of failures and part successes, past and of more recent date, will help forward the selection of the essentials necessary for the ideal design.

The rotary valve, for technical reasons and because of the sound principle upon which it is based, is in the not far distant future bound to show general superiority over systems employing reciprocating parts. If knowledge of the wealth of experimental work and of actual performances can be more widely distributed, it will dispel many of the misunderstandings which exist in regard to the merits or otherwise of the rotary valve and should provide a nucleus of information which will prove of value in educating public opinion. The education of public opinion is necessary because without it new ideas are of little commercial value, however good they may be technically.

Talking of British science and invention, Lord Riverdale only recently said:

> "It would be easy to point out a dozen or more first-class inventions that have been invented in this country; nobody would take any interest in them and they have been bought by the Germans and either used as they were or applied to some research they were doing and for which they have afterwards obtained very substantial results."

The significant fact is that we have many times failed to apply modern scientific discovery quickly.

It is of paramount importance that, when a stage of general popularity is reached, the British industry shall be at the head of the rotary valve epoch and not at the tail of its rivals.

It is well to point out that in compiling the information forming the subject matter of the following chapters a genuine difficulty has been experienced in tracing authentic reasons for the discontinuance of certain productions incorporating the rotary valve. The designs in some cases have shown considerable merit and needed little more than perseverance to overcome the detail mechanical difficulties. Undoubtedly in some instances the financial or commercial aspect has come into play, applying a brake to the progress of a promising principle and preventing the natural growth of ideas, which might long ago have led to outstanding engineering products.

This view is confirmed by a statement once made by Sir Dugald

Clerk, F.R.S., F.C.S., father of British gas engines, at a meeting of the Institution of Automobile Engineers:*

> "I am sorry to say that many of the best things I have ever designed have been ruthlessly set aside by my brother directors, on what they falsely called economic considerations. It seems rather ridiculous, gentlemen, that a scientific man, who produces improvements, should be hampered by other directors, who insist on earning dividends for the shareholders. I like the dividend when I get it, but I do not like my experiments being stopped."

These words convey bluntly what befalls some inventions. However, it is more for the technical interests and the historic facts that this book has been written. Sometimes an invention is before its time, as regards both the quality of material and the precision methods of production available. A relapse then occurs, but after the passage of time progress in the art enables the idea to reassert itself, and the threads are picked up by others fortunate in the possession of more advanced materials and tools.

Full credit can be given to the few who, undaunted by early setbacks, are now in a position to show rotary valve engines on or near a production basis. These engines, of great diversity in design, are well on the way to success.

The present improvements have resulted directly from the knowledge of what has gone before, and likely progress in the future will similarly depend on the dissemination of the information and experience gained under the active stimulant of war.

* Vide *Proc. I.A.E.*, 1911–1912.

CHAPTER I

HISTORICAL FOREWORD

LITTLE more than 70 years ago an engineer named Otto, later of four-cycle fame, exploded a scientific bombshell over the continent of Europe, the repercussions of which reached the workshops large and small of numerous steam-engine makers. The owners of these workshops attempted to prevent the invention of the gas engine from interfering with their profitable business. But interest in Otto's invention soon reached Britain, and in the early days of 1875 Mr. F. W. Crossley read a paper before the Institution of Mechanical Engineers in London entitled "Otto and Langen's Atmospheric Gas Engine". This event started the wheels moving for the production of some of the first internal combustion engines to be made in this country.

About the same period Karl Benz was labouring on an invention akin to this in his small workshop at Mannheim, Germany, and it was in the year 1878 that he produced his first engine, a two-stroke engine of one horse-power. In the same year Dugald Clerk constructed a gas engine of some three horse-power. This is how Mr. Clerk has described the engine he made at that period:

> "It had nothing in it but slide valves; that is, the whole functions of the engine were performed by slide valves, so that I had some experience of sliding surfaces and from my experience I did not like them at all. . . ."

Another of Dugald Clerk's prototype engines was built in 1881, and can, or could until recently, be seen in the Museum at Munich. It is interesting to know that these experimental internal combustion engines were produced several years before Gottlieb Daimler was granted his first patent for a gas engine with tube ignition in the year 1883. It is more than probable that all of these historical engines were fitted with reciprocating slide valves, following steam-engine practice of the period. It will be gathered from the description in a subsequent chapter, which makes reference to the Crossley gas engines of 1886 to 1902, that during those years the reciprocating slide was replaced by a rotor which revolved at a uniform speed in one direction and thereby established for the first time the principle of a valve gear free from inertia stresses. By the end of 1902 the contemporary poppet-valve had proved itself so reliable in practice that it swept the field and was adopted almost universally

for stationary gas engines of all powers and also on petrol engines running at much greater speeds.

This is not to say that experimental work with rotary valves was entirely dormant, for it is known that Mr. E. W. Lewis at Coventry about the year 1908 produced a four-cylinder engine with a rotary valve. Records also show that Mr. C. W. Paget, then Works Manager at the Midland Railway Works, Derby, had applied the principle to a main-line steam locomotive, not without some success. Also the National Gas Engine Company at Ashton-under-Lyne was carrying out research work under the able guidance of Mr. Dugald Clerk, who, at a meeting of the Institution of Automobile Engineers in 1911, presented this interesting reminiscence:

> "Some time ago the National Gas and Oil Engine Company Ltd. tried a number of very interesting experiments which bear out what Mr. Coffin tells us, that is, that in America they are using rotating valves so as to get rid of the reciprocation. It is quite praiseworthy to get rid of the reciprocation if you can—notwithstanding I see Mr. Knight here—and I will tell you what we did in the National Gas Engine Company's works some years ago. We built a small engine with a centre plug valve and we surrounded it with a casing, and that casing we surrounded with exhaust gases instead of with water. If the latter is used of course the plug expands and you very soon come to a deadlock. The consequence is that in order to start your engine at all you must begin with a loose rotating plug which must be so loose that when it becomes hot it will not jam in the cylinder. That puts any parallel plug running in a cold jacket out of the question. What we did was to make both plug and jacket expand at the same rate, and, therefore, we got both hot. That worked well and one engine worked in the National Company's works for nearly two years and ran beautifully. The speed of revolution was very high, about one thousand revolutions per minute, and we had no difficulty whatever, but we had such a good sale for the ordinary engine and the ordinary engine was so fool-proof that for commercial reasons, and also for economic reasons, it was not put on the market."

So sliding valves were abandoned for a number of years and the poppet reigned supreme. Then, in the year 1908,* public opinion received a shock, this time from America, when we find the Daimler Company of Coventry taking up Mr. C. Y. Knight's engine with "two slides" per cylinder—nearly as bad as Mr. Dugald Clerk's "all slides", which he did not like.

* C. Y. Knight's British Patent No. 12,355 is dated June 1908.

Conservative engineers at that time said that this was all wrong—slides could not work, they should not work, they must not work, but if they did work, then they ought not to slide to and fro, but round and round. In spite of the critics Daimlers prospered, and the invention of the Burt McCollum single sleeve engine added impetus to the movement towards the adoption of sliding valves, although the house of Argyll Ltd., Alexandria, N.B., who had adopted the invention and expended a considerable sum on its development, crumpled in the process. These two outstanding successes stimulated designers both in Britain and in U.S.A. to produce valves without the complication of reciprocating parts. It is noteworthy that during the six months following the granting of the British patent to C. Y. Knight for the double sleeve valve engine, there were no fewer than 21 patents for valves of rotary type taken out in this country—eight of these covered rotating liners, nine valves of cylindrical form and the remaining four valves of disk type. Yet during the six months period preceding the acceptance of the Knight patent there was not a single British patent issued covering a rotary valve of any description whatever. When members of the American Society of Automotive Engineers made a visit of inspection to Europe in 1911 and were invited to attend a general meeting of the Institution of Automobile Engineers in London, Mr. Howard E. Coffin (Past President of the A.S.A.E.) joined in the discussions on the red-hot subject of valve gear with these remarks:

> "Of course, no engineering discussion would just now be complete without some mention of special motor valves. The Knight motor you know. It is now being adopted by four makers in the States, and apparently with success. There are several American valve mechanisms which seem to promise well. Every one is of a rotary disk or rotary valve type. It seems to be the American creed that if the poppet-valve is to be dropped, it would be a mistake to replace it with another reciprocating mechanism—sleeve or otherwise. Hence a concentration upon that type of valve action which may be accomplished by a mechanism free from reciprocating parts. Some of these constructions have shown a fine performance on the road and the difficulties would seem small when compared with those overcome by Mr. Knight earlier in the art."

What became of these fine performances in the United States of America it is not here possible to say, but no design survived so well or so long as the Knight and Burt McCollum sleeves.

Contemporary with the Knight engine, there appeared at the Olympia Show in 1911 a car fitted with a four-cylinder engine

by Messrs. A. Darracq and Co. Ltd., incorporating a single cylindrical valve extending the full length of the engine. At about the same time, the Itala Co. of Turin marketed a four-cylinder car with vertical rotary valves, one valve per pair of cylinders. Both of these motor-cars enjoyed a certain measure of success on the road, but the models had become extinct before the commencement of the First World War. Little in the way of rotary valves has been marketed since then, but recently several ideas have matured and are in active course of experiment or manufacture and, as far as information is available, will be fully described in subsequent chapters.

CHAPTER II

VALVE DEFINITIONS AND PRINCIPLES OF DESIGN

MECHANICAL valves—using the term in the widest sense—are by far the most important part of any steam or combustion power unit, for upon them the functioning and efficiency of the whole engine depend. The rotary valve is considered by some to be the most advanced and simple of all. The following pages will treat on their development without encroaching on the general design of the engine as a whole, the construction and theory of which have been covered by standard works.*

Before undertaking a critical and impartial examination of a number of historic and proposed designs of rotary valves, it is desirable to consider briefly the present state of knowledge of some ordinary mechanical devices and engineering principles of proved merit in universal use. Too much stress cannot be laid on the fact that good practice is always the result of adhering to fundamental laws, and nowhere is this more in evidence than in the early development of the steam engine and hydraulic machinery.

Broadly, there are two groups of mechanical valves used by engineers, and these may be classified under the main headings—LIFT-TYPE and SLIDING-TYPE. Sliding valves are sometimes termed surface valves. Both types have been used on internal combustion engines and steam power units for diverse purposes but to perform the same function, i.e. opening or closing a duct for the passage of gas or liquid. The selection of one form or another for any purpose depends upon many factors, chiefly convenience and the required speed of operation.

Description of a Rotary Valve

The domestic turn cock for water or gas shown in Fig. 1 is an elementary type of rotary valve. It is, further, an example of one which will function with either a semi or a full rotary-movement. If turned through an angle of 90° in either direction the port will be changed from a closed to an open position. Alternatively, the rotor can be operated always in one direction and the cycle "closed—open" will be repeated. In simple form therefore it is possible to define a rotary valve as comprising an outer casing or body, an

* Vide *Automobile and Aircraft Engines*, by A. S. Judge, and *High Speed Combustion Engines*, by P. M. Heldt.

inner element provided with a port for opening and closing a passageway and capable of being rotated by a handle or lever. This is a general description and rotary valves may be sub-classified into groups according to the form of the moving element, the disposi-
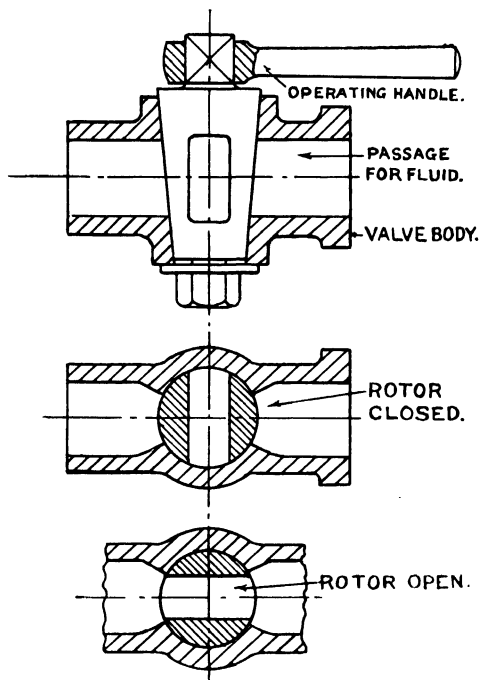
FIG. 1.
Domestic turn cock. An elementary form of rotary valve.

tion of the ports and the kind of rotary motion, oscillatory or continuous. It will be clear from the following definitions that all forms of rotary valves are of sliding surface type. It is also important to note that friction is inherent to their operation.

Lift Valve

The lift valve is so named because it functions by lifting entirely off its seat. In motor-car practice it is referred to as a poppet-valve. Fig. 2 shows a lift valve in the open position. The effective area of the opening depends upon the mean circumference of the annulus and the distance L, with a strict limitation by the available area of the throat minus the cross sectional area of the valve stem. The

ordinary safety valve on a steam boiler falls within the definition, but in this case the seating of the valve is usually made flat and in
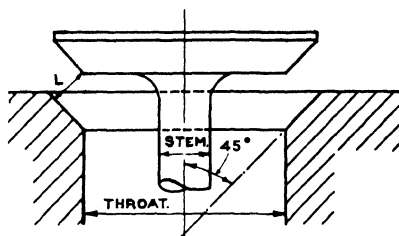
FIG. 2.
Lift or poppet-valve with seating at 45° showing effective opening L.

this respect is different from the poppet-valve of an I.C. engine, which in general has a conical seating with an included angle of 90°.

Sliding Valves

The sliding valve is so called because one surface slides upon another. One or both surfaces are provided with ports or orifices which are open or closed according to the relative position of the sliding members. The effective area is the area of that portion of the port which is uncovered. The group definition embraces valves with surfaces which may be flat, curved or cylindrical and moving either to and fro or progressing with a rotary motion. It therefore includes all rotary and semi-rotary valves. It is convenient to sub-divide sliding valves into two classes according to the type of movement given to the valve. *Reciprocating Slide Valves* are
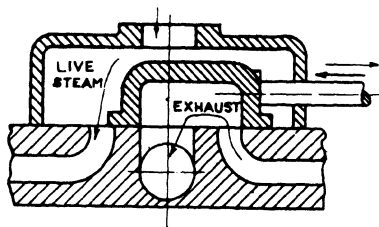
FIG. 3.
Slide or surface valve as used on a double acting steam-engine.

valves having a reciprocating motion, such as the well-known D slide valve of a steam engine which is shown in Fig. 3. This

valve is moved to and fro by the operation of an eccentric or crank. The Knight double sleeve valve used by the Daimler Co. falls in the same reciprocating class. The single sleeve valve used by the Argyll Co. has a combined reciprocating and semi-rotary motion, so it cannot be strictly termed a reciprocating valve although it is of the sliding surface type. *Rotating Slide Valves.* Valves which are circular and move continuously in one direction are strictly rotary valves. These may take the form of disks, cones or cylinders of solid or tubular section. A valve component which moves through only part of a revolution or combined with a to and fro movement does not fall strictly in this category, but may be distinguished by the nomenclature semi-rotary valve. The rotating member in all the above cases is generally referred to as a rotor. A rotor, which moves only in one direction and at approximately uniform speed, provides the engineer with a type of valve which is unique in that it can be balanced so that its movement produces no inertia stresses. In consequence there are no limitations as to mass or velocity, except for stresses introduced by centrifugal force, which are in a plane above discussion at the moment.

Types of Rotors

Basic types of rotary valves, of disk, cone and cylindrical form, are shown diagrammatically in Figs. 4 and 5. In Fig. 4 the revolving disk is arranged in proximity with the cylinder head, in the way it would be applied to an I.C. engine. The surfaces of both valve and head must, of course, be highly finished, and gas-tightness is assured by the pressure within the cylinder holding the disk during the compression and explosion strokes in close contact with the underside of the top face of the cylinder. The positions of the ports are so arranged that they will provide the correct timing, there being two ports in the cylinder head, one leading to the intake and the other to the exhaust pipe.

As a variation in design, the disk is sometimes replaced by a cone as indicated in the detail, instead of being made with a plane surface. The cylinder head is then formed to fit the cone instead of being machined flat.

In the applications where this type of disk rotor is used in a steam engine for the admission of live steam to the cylinder, or in an engine operated by compressed air, the position of the rotor is generally inverted as shown in Fig. 6; that is to say, the valve is placed within a steam or air chest above the cylinder head instead of beneath it, in order that the self-sealing feature can be usefully employed in maintaining a gas-tight joint between the rotating member and the cylinder against the live steam during the exhaust
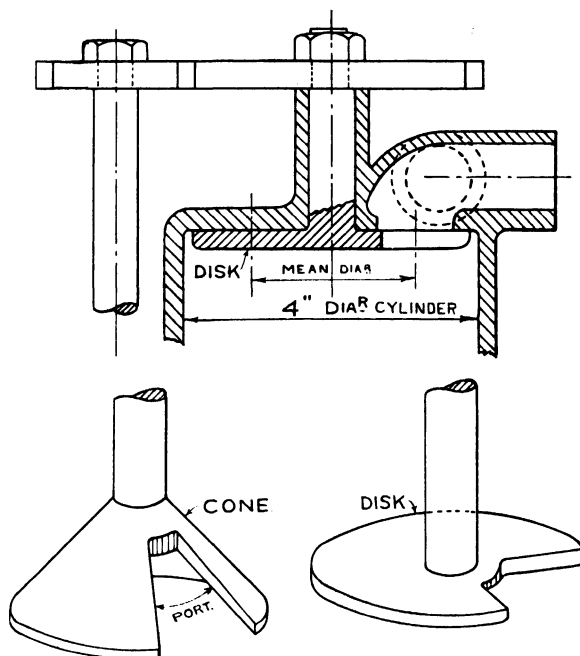
FIG. 4.
Disk and cone rotary valves, as applied to internal combustion engines.
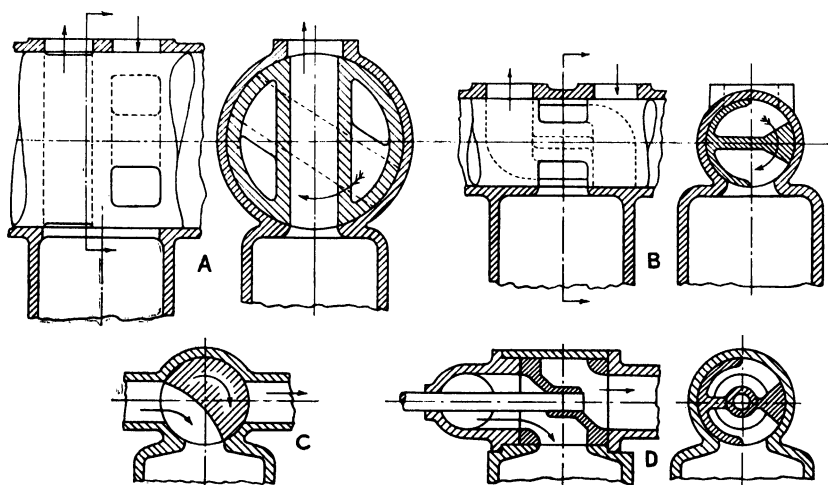
FIG. 5.
Basic types of cylindrical rotors. A rotates at one-quarter engine speed. B, C, and D rotate at one-half engine speed. All have the same chordal-width of port and approximately the same surface velocity.

stroke. This requirement is a first essential with any type of valve used in connection with a prime mover operated by steam or air under pressure. The self-sealing feature of the disk valve is inherent and of great simplicity, although, as will be explained later, this form of valve is not ideal from the viewpoint of mechanical friction. In the case of a rotor of cylindrical form there is no similar sealing action, and, in consequence, some extra device has to be introduced between the rotor and the port-opening in the cylinder to ensure a gas-tight seal.

Fig. 5 shows in diagrammatic form alternative arrangements of cylindrical rotors A, B, C and D. The arrangements are all suitable for general application to four-stroke internal combustion
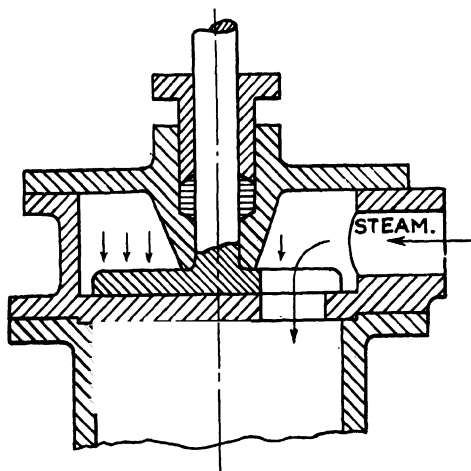
FIG. 6.
Rotary disk valve arranged for steam or compressed air engine.

engines. It is fundamental that the type of rotor marked A must be geared to rotate at one quarter of the engine speed. The others, marked B, C and D, must rotate at half of the engine speed in order to satisfy the requirement of a four cycle engine with one exhaust and one admission stroke during two revolutions of the engine. The rotors are shown in elementary form without special sealing devices and without any cooling jackets.

SLEEVE ROTORS. The basic sleeve type of rotor can be put in a special class because it is distinctive in two important particulars. First, the form is tubular, and secondly, it encircles the piston. Therefore it forms a working liner inside the cylinder barrel. Fig. 7 shows a sleeve arranged to revolve in one direction at a uniform speed. It is operated by a worm and worm wheel at the lower end,

and it is driven by suitable gearing from the crankshaft. Two ducts are provided in the combustion head, one for the inlet and the other for the exhaust. A single port is cut in the wall of the sleeve and this communicates with the inlet and exhaust, each in turn, as the sleeve rotates. The speed of rotation, for a conventional four-stroke engine, is half the speed of the engine. However, if the sleeve is provided with two ports diametrically opposite each other,
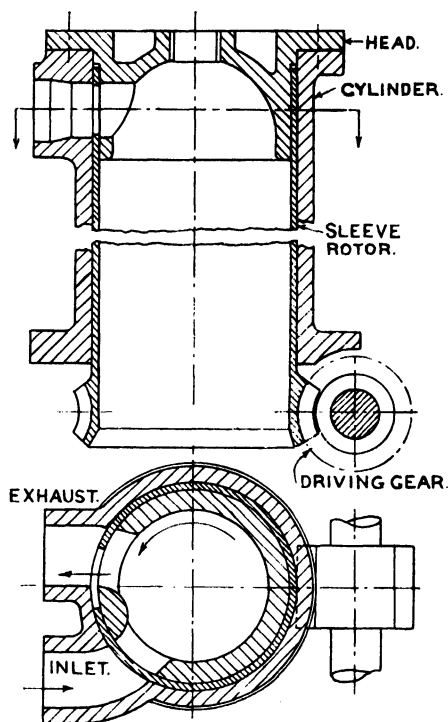
FIG. 7.
Sleeve type of rotor arranged inside the cylinder and to rotate at uniform speed in one direction.

the speed of the sleeve will in consequence have to turn at one-quarter of the engine speed.

A form of valve operating with an unusual movement is that shown in Fig. 8, which illustrates the principle of the Burt McCollum single sleeve valve. The rotor is tubular and is similarly located between the piston and the cylinder wall. The upper end has formed in it ports of special shape. It is driven by a crank with a sliding plunger and a hinge pin. Consequently the sleeve is rotated about its axis through part of a revolution, the actual amount depending on the throw of the crank, and at the same time the

sleeve is given an up and down motion, thus bringing the ports opposite the inlet and exhaust passages alternately in correct
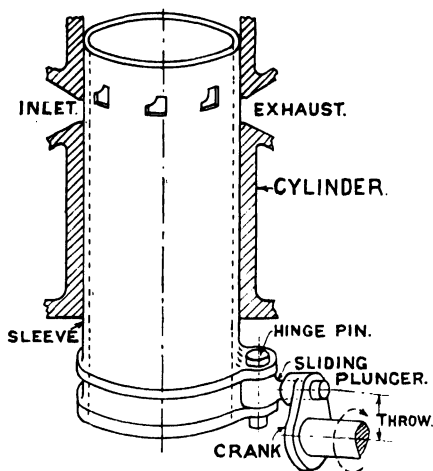
FIG. 8.
Semi-rotary sleeve rotor—generally known as the Burt McCollum single sleeve valve.

sequence. The vertical travel of the sleeve is equal to twice the throw of the crank.

Advantages of Uniform Rotary Motion

All reciprocating valves of whatever form, poppet, sleeve or semi-rotary, are subject to alternating inertia stresses which impose a definite requirement for the strength of the components, and a limit on the permissible speed of the engine. The inertia stresses vary as the square of the speed according to the formula—Energy $= M.v^2/2g$, where M is the mass of the moving part, v equals the velocity in feet per second and g equals the acceleration due to gravity. The advantage of rotary motion is unlikely to be long neglected by automobile and aero engineers, who are persistently aiming at increased rates of revolution in the quest for greater power for the same weight or from the same cubic capacity of cylinders.

The primary advantage of absence of vibration is accompanied by a secondary advantage, namely silence, as the conditions of circular motion are also most favourable for quiet operation with minimum vibration. Another feature of great value is the fact that rotary valves do not require any periodic adjustment throughout

the whole life of the engine. The labour and time necessary to adjust the multitude of tappets in a poppet-valve engine on first assembly are entirely saved.

The degree of silence under operating conditions should, on theoretical grounds, surpass the high order of quiet operation now attained in practice with reciprocating parts; this is by recourse to the finest workmanship. Furthermore, a poppet-valve engine is always associated with a high frequency vibration which cannot be fully eliminated.

Difficulties to be Overcome

To gain the desirable advantages of absence of inertia stresses and silent operation, qualities inherently possible by adopting the rotary principle, the designer is faced with great practical difficulties, which have to be overcome by attention to minute detail and a rigid application of correct engineering principles. Mr. Laurence Pomeroy, technical editor of *The Motor*, has listed some of the difficulties and his list bears repetition:

(a) Maintaining a clearance between the running parts sufficiently close to avoid excessive gas leakage and loss of compression at low speeds and sufficiently wide to avoid seizure following upon distortion at high speeds and loads.

(b) A supply of oil to the working faces which will be adequate to prevent seizure under the most severe conditions and will not cause plug oiling or excessive oil consumption.

(c) The isolation of each valve from the effects of inter-cylinder distortion on a monobloc in-line engine.

(d) A provision of a drive or drives to the valve or valves which will not be exceptionally cumbersome or costly.

(e) An avoidance of expensive items of construction, which would prevent successful marketing even if the performance was technically meritorious.

The major problems involved will be treated under three broad headings: (I) Sealing devices to ensure gas tightness. (II) Friction, lubrication and power losses. (III) Expansion and distortion due to temperature changes. Questions of valve area, dimensions, proportions and the application of the valve to the engine will be dealt with in the next chapter. It is proposed to consider how, by a scientific approach, analogous difficulties have been overcome in other fields of engineering and later to see how far the essentials have been met in some of the early proposals and typical engine designs reviewed in subsequent pages.

(I) *SEALING DEVICES TO ENSURE GAS TIGHTNESS*

This problem is as old as the steam engine, and a sealing device to ensure gas tightness against high pressure between two moving parts has been developed by the expedient of employing the pressure of the gas or water to contribute to its own seal. The D slide valve of the steam engine cylinder as shown in Fig. 3 is a representative example. It will be noted that the pressure of the steam confined in the steam chest holds the sliding member on to the machined face, through which the steam, by the medium of ports, passes on its way to the working cylinder. The same principle is illustrated
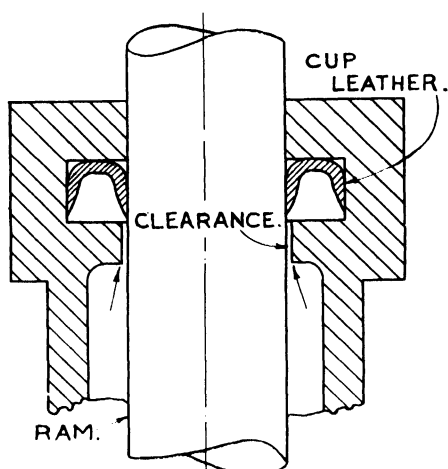
FIG. 9.
Hydraulic cup leather invented by Bramah for sealing a ram.

in the cup leather invented by Bramah, Fig. 9. The circular cup leather or flexible packing ring is pressed into contact with the ram on one side and the guide on the other side by the water which is allowed to pass to the packing ring through the clearance, and the result is that the greater the water pressure, the more efficient does the seal become.

The principle underlying the action of the ordinary piston compression ring is analogous, although, at first sight, not quite so obvious. It is an essential feature that the pressure within the cylinder should have access to the space behind the ring, so that the latter is expanded by a force in addition to the normal outward spring of the ring. It will be noticed in Fig. 10, where the clearances are shown somewhat exaggerated, that the ring is forced on to the lower face of the piston groove, and this allows the proper admission

of pressure to the inside of the ring, thereby effecting the seal between the outside of the ring and the cylinder wall. It is, of course, necessary for the adjacent surfaces of limited area to have a fine degree of finish. In practice the clearances are made small so that the pressure is wiredrawn to some extent to prevent excessive wear.

In the well-known Knight sliding-sleeve engine the sealing of the ports during the compression and explosion strokes is effected by the same method, the pressure itself within the cylinder being employed to maintain the tightness of the port apertures. The pressure admitted behind the junk ring, Fig. 11, tends to keep it
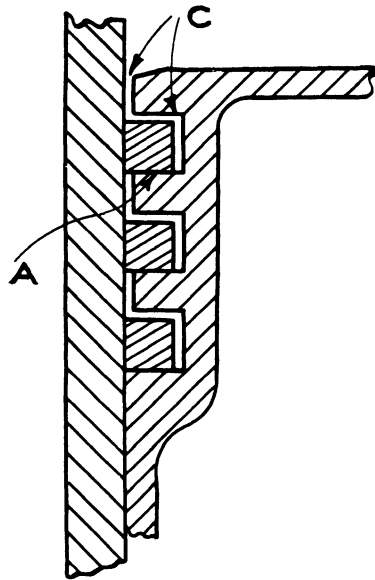
FIG. 10.
Compression ring for ordinary engine piston showing sealing surfaces A and exaggerated clearances C.

home on the inner face of the inside sleeve, and at no time does either the compression pressure or, what is of more vital importance, the explosion pressure ever find admission to joints purely dependent upon accuracy of machining. It must be noted that clearances are necessary, but only in the right places.

Here then is a principle of the highest importance which cannot be overrated and one that has withstood the test of time and proved itself sound as well as adaptable and for which there appears to be no better alternative. The higher the pressure, the greater the force applied to ensure the seal. It must be admitted that the higher the pressure the greater the friction, but the pressure asso-

ciated with I.C. engines is in the ordinary course of things cyclical, and at times negative in value, so that the mechanical power losses from friction, as explained later, are computed on the basis of the average pressure only. Such pressure values are not of a high order, even in Diesel or boosted petrol engines, when compared with the working pressures common in hydraulic and other machinery. Sufficient has been said at this juncture to stress the
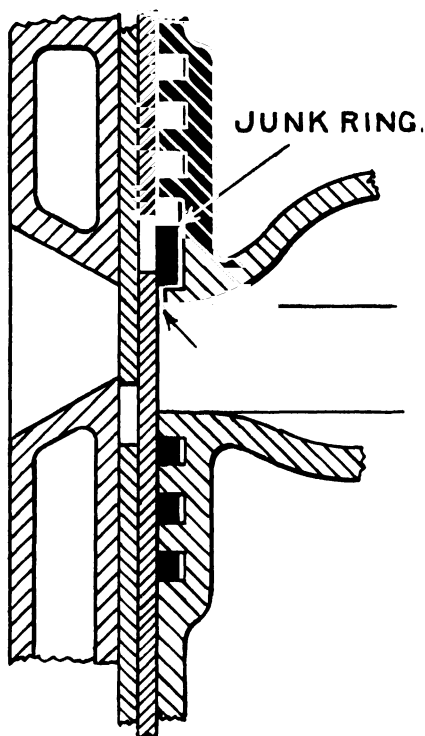
FIG. 11.
Knight double reciprocating sleeve valve with junk ring for sealing the port opening of the inner sleeve.

need for the employment of the self-sealing principle. The requirement has been carried out in a number of ways on some of the rotary valve engines described. In cases where the principle has been neglected, the designs have not generally survived the experimental stage.

The inherent mode of sealing for a valve of disk form without the addition of extra parts has been described and applies equally to a conical valve, but before the same principle can be extended and adapted to a rotor of cylindrical form it is necessary to introduce

some special device. A number of schemes are shown diagrammatically in Fig. 12 (A), (B), (C) and (D). A careful consideration of these diagrams, at this point, will help to clarify actual examples as applied to typical engines shown later, where the advantages and defects will be more fully explained.

In scheme (A) the cylinder is provided with a liner or sleeve, wherein the head is closed except for a port-opening of similar size to that provided in the rotor. The liner is arranged with a
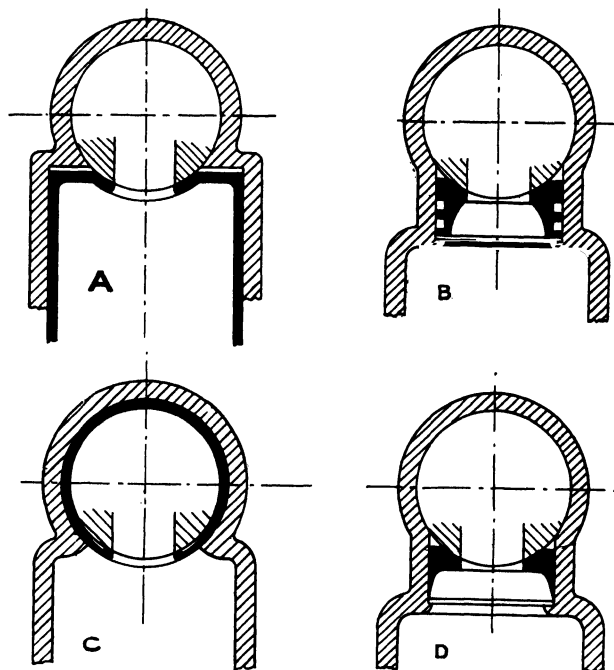
FIG. 12.
Basic forms of sealing devices for cylindrical rotors. A—Liner with closed head and port orifice. B—Sealing piston with compression rings. C—Resilient bush with lipped edge round the port. D—Sealing piston with feather edge to seal the diameter.

certain amount of vertical freedom in the cylinder block, and the upper end is machined to a radius which conforms to the radius of the rotor. During operating conditions the pressure within the cylinder forces the liner upwards against the rotor, thereby effecting such close contact between the two surfaces as is necessary to produce a gas-tight seal. It will be noted that the load between the running surfaces is equal to the product of the gas pressure in lb. per sq. in. within the cylinder multiplied by the area of the engine bore in sq. in. less the area of the port.

In the second example, as shown in (B), the cylinder head is provided with a short piston fitted with one or more ordinary compression rings, and this piston is formed on the upper surface to fit the rotor in the same manner as described in scheme (A). Similarly the crown of the sealing piston is pierced with an aperture to mate with the port of the rotor. It will be observed that in this scheme a second path of possible leakage is introduced by way of the compression rings, and as there can be no actual working of these parts, and consequently no running-in process, the fit must be near perfection in the first instance. A great advantage, however, accrues from the arrangement, as the sealing piston can be made with an area considerably less than that of the engine bore, and, in consequence, the load on the rubbing surface is less and the resulting friction is proportionally reduced.

The third scheme (C) is one in which the natural spring of the material encircling the rotor is brought into use. In this case the cylinder head is bushed with a comparatively thin phosphor bronze tube or like material, and the edge round the port orifice is lipped upwards to provide an increased local contact pressure, which at all times helps to produce a gas-tight joint. During the compression and explosion strokes the load on the seal is enhanced due to the pressure within the cylinder tending to press the resilient material of the bush into closer contact with the rotor, thus to some extent employing the self-sealing principle.

In the arrangement indicated in diagram (D) the method of sealing, at the surface in contact between the top of the ported piston and the rotor, is exactly the same as that described for (B), but an alternative is introduced to take the place of the compression rings. The lower end of the piston is turned to a feather-edge and spun outwards to a slightly larger diameter than that of the bore into which it fits. The result of this is that when lightly forced into position the fit remains perfect at more or less all temperatures and the resilient section of the material at the feather-edge allows the pressure within the cylinder to act on the feather-edge in accordance with the self-sealing principle. In some constructions the feather-edge is produced on a separate component, and in these circumstances it becomes what may be described as a metallic obturator ring, somewhat similar in form to the Bramah cup-ring used in hydraulic machinery.

There are, of course, a number of variants based on the above four typical examples and also refinements in detail to suit particular applications. An additional feature is sometimes introduced in the form of a partly balanced pressure on the valve with the object of avoiding excessive load on the rubbing surface. An alternative method to pressure balancing has been developed by one constructor with great success, and he gains the same result by a system of applied leverage. By these means the load on the sealing surface

may be reduced to any extent, the levers being proportioned in the design stage to give minimum friction consistent with adequate pressure for sealing.

Most of these pressure-balancing or pressure-reducing devices show great ingenuity, but not all of them are of practical value. Nevertheless the principle is so important that it can be stated that unless it is incorporated in the design, failure will be the result.

In sleeve valve engines a number of compression rings of the type usually fitted to the main pistons are employed for sealing the gas pressure between the inside diameter of the sleeve and the cylinder head. This form of head is often called a junk head, and the sealing rings are fitted to grooves machined on that part of the
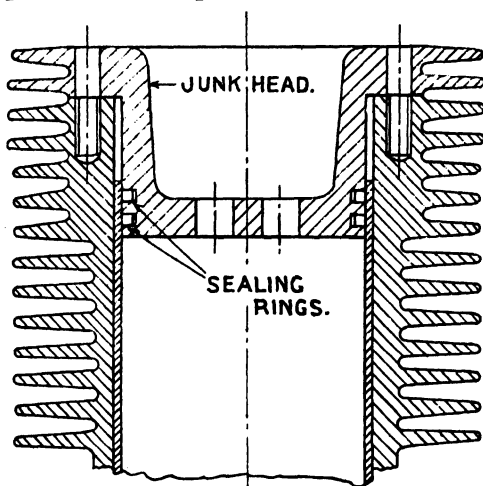
FIG. 13.
Air-cooled head of aero-engine with compression rings for sealing the sleeve rotor.

head which is encircled by the sleeve. A typical assembly of an air-cooled junk head, as used on aircraft engines with semi-rotary sleeve valves, is shown in Fig. 13. It will be noticed that in this arrangement no friction will result from the pressure within the cylinder except that due to the rings and side thrust from the connecting rod. During the firing and compression strokes the port apertures in the sleeve are buried behind the rings; in consequence no further sealing device is necessary. This is an outstanding advantage of the semi-rotary sleeve, but it cannot apply to a sleeve which revolves without oscillatory movement. For instance, in a design such as shown in Fig. 7 on a previous page, the edges of the ports are exposed to the cylinder pressure; sealing of the aperture is therefore dependent on the working clearance between the outside of the sleeve and the inside surface of the cylinder barrel.

(II) FRICTION, LUBRICATION AND POWER LOSSES

The sliding valve with few exceptions involves the movement of one loaded surface upon another, and thus produces friction. The lift valve of poppet type has sliding contact on the stem only, and, as this is comparatively small in diameter, it carries little transverse load, and therefore the loss from friction, excluding that due to the operating mechanism, is negligible. Moreover, the need for lubrication in the latter case is of little importance, in spite of the fact that the centre of the head of an exhaust valve may reach a temperature of 760°C. or more. In connection with sliding surfaces, however, it is necessary to give careful consideration to lubrication in order to reduce the friction to a minimum. This is a problem of great magnitude, calling for all possible ingenuity on the
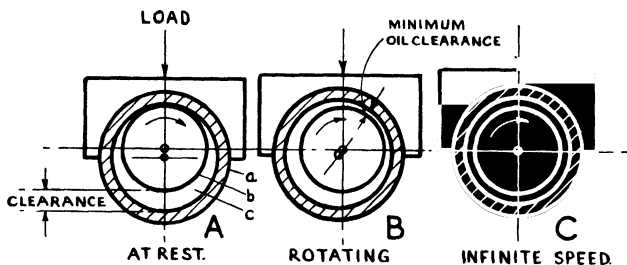
FIG. 14.
The lubrication of a running bearing. Positions of shaft at rest, rotating, and at infinite speed. (A) Bush. (B) Journal of shaft. (C) Clearance space.

part of the designer if his rotary valve gear is to survive the test of time.

This is not the point at which to lay down the best scheme of lubrication for a rotary valve, but it is desirable to point out some of the steps found necessary for efficiency in two classic examples of sliding friction encountered in ordinary engineering practice, namely the JOURNAL bearing and the THRUST bearing, and the conditions that have been proved essential to promote the best results. These remarks apply to bearings working at quite a moderate temperature, certainly lower than that which obtains during the functioning of a rotary valve in an I.C. engine or a high-pressure steam engine. The difficulties to be overcome in the lubrication of rotary valves working at high temperatures will be obviously greater.

LUBRICATION OF A JOURNAL BEARING. The journal or shaft bearing is in some degree analogous to the rotary valve of cylindrical form. The general established theory of the lubrication of a

running shaft will be reviewed. Fig. 14 illustrates a typical case. The bush surrounds a journal, and the clearance space, which may be very small, is, under normal working conditions, filled with oil.

When the shaft is stationary the loaded bush rests on the top of the journal, as in position A. If the shaft is rotated in such a way that oil is drawn into the space between the shaft and the bush, the latter is displaced from the position it occupied when at rest. As a result of several factors such as adhesion, rubbing speed, viscosity and the wedge shape of the annular space, the lubricant is drawn in the direction of rotation of the shaft to the point of minimum clearance, in accordance with the laws of hydrodynamics. The bearing lifts out of contact with the shaft as shown at B and finally—if the speed is sufficient—takes up the theoretical concentric
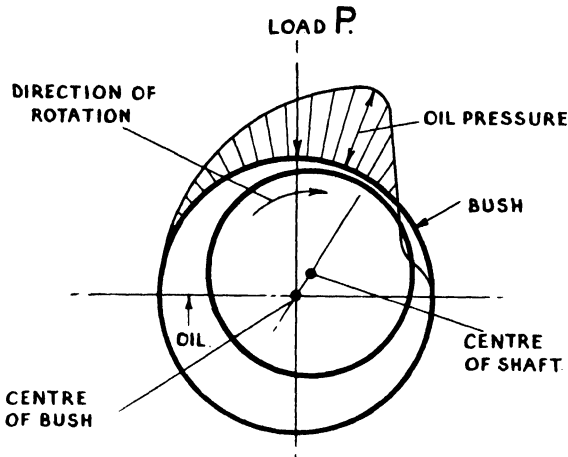
FIG. 15.
Polar diagram of oil pressure in a running bearing. (*E. Falz.*)

position C, floating centrally on the oil, and therefore free from any metallic contact with the journal.

The normal equilibrium position is more clearly shown in Fig. 15, where the shaft, carrying load P, revolves in the direction of the arrow. This figure also includes a polar diagram (E. Falz) representing the variation of pressure in the oil film. The shaft, surrounded with oil, and not in metallic contact, is in the ideal condition where the materials of construction have little influence on the friction when once the shaft is rotating. The oil wedge theory does not hold entirely if either the shaft or the bearing has an interrupted surface, and, in consequence, any kind of port or aperture becomes an adverse factor to successful lubrication.

LUBRICATION OF A THRUST BEARING. A thrust bearing is approximately the equivalent of one flat disk heavily loaded and

rotating upon another. The design of a reliable device proved to be one of the most elusive problems met with in the early days of ship propulsion, and it is still a problem not entirely without difficulties in modern steam turbines. The invention of M. A. G. Michell, of Australia, about the year 1913, provided one solution. The design is based upon the theory of the oil wedge, and the good results are achieved by a ring of tilting pads as shown diagrammatically in Fig. 16, and in a typical assembly Fig. 17. The disk
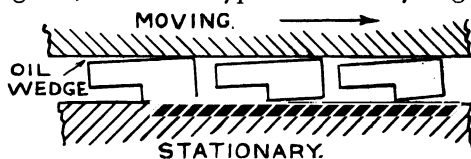
FIG. 16.
Position taken up by the pads in a Michell thrust bearing showing the oil wedge exaggerated.
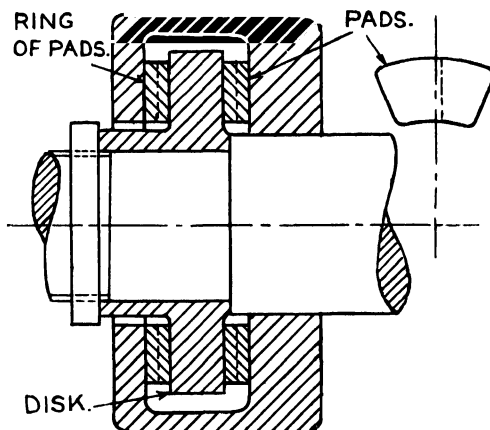
FIG. 17.
Double thrust bearing of Michell construction. The detail shape of a pad is shown on the right.

revolves between two rings of pads, and in this specific arrangement thrust may be taken in either direction. This is without a doubt one of the most notable detail inventions of our time, and has had a more profound effect on many types of machines than any other single invention of general application. The theory of the oil wedge has been applied in a somewhat different manner by Mr. C. C. Pounder in connection with marine thrust bearings* used by Messrs. Harland and Wolf, Ltd., Belfast. A section through the thrust block is shown in Fig. 18. A single split hollow thrust disk

* "Types of Propelling Machinery", by C. C. Pounder. *Proc. Inst. of Mech. Engineers*, 1943, vol. 150, No. 2.
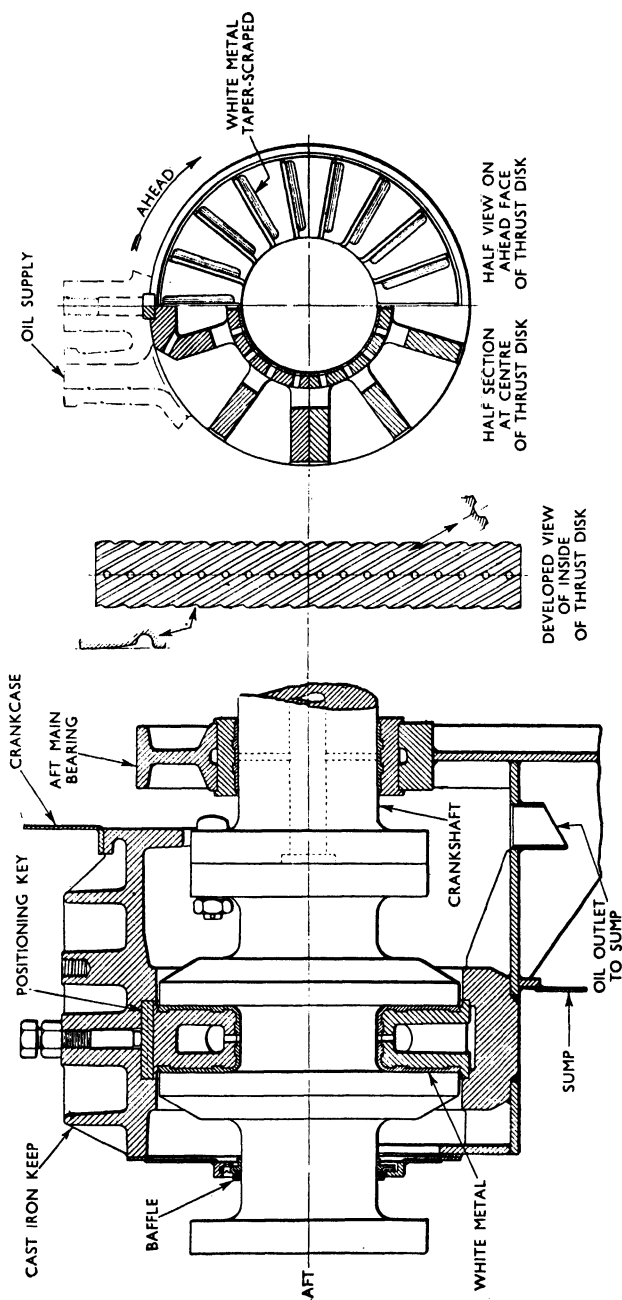
Fig. 18.
Marine thrust bearing developed by Messrs. Harland and Wolf, Ltd., Belfast.

is arranged between two collars. The bearing surface on each face of the disk is divided by radial oil grooves into sections, the mean width of which is, at the most, three-quarters of the radial length. These radial grooves are continuations of helical grooves cut in the bore of the thrust disk. Oil is supplied to the helical grooves through holes drilled into the hollow casting of the disk, which serves as an oil reservoir. The radial grooves are carefully taper scraped at one side, the oil being induced up the taper as the thrust collar rotates, thus ensuring a continuous oil film for sustaining the thrust load. The coefficient of friction claimed for this bearing, depending upon viscosity, pressure and speed, lies between 0·0035 and 0·0075 with loads of approximately 1700 lb. per sq. in. The coefficient is smallest with high speeds and pressures and warm fluid oil.

It will be obvious that neither of these arrangements for the lubrication of thrust bearings can be applied unaltered to a rotary disk-valve for an I.C. engine. Here, the minimum quantity of oil is a requisite if fouling of sparking plugs and smoking of the exhaust are to be avoided. Prominence has been given to the above classic examples of journal and thrust bearings in order to stress the importance of analytical examination of any proposed system of lubrication and to indicate the serious nature of the problem. It is never safe to rely on haphazard methods for the solution of difficulties which require the most profound consideration.

LUBRICATION OF SLIDING SURFACES. In order to show that special problems of sliding friction can be, and have been, solved, it is only necessary to quote such examples as the Bristol single sleeve valve air-cooled radial engine and the Knight double sleeve automobile motor. Both these types of sleeve valve, in the early days, were strongly criticized by conservative engineers as impossible propositions, but by careful design have since proved eminently satisfactory. Mr. C. Y. Knight thus described the oiling system of his engine* at the time of its introduction into this country by the British Daimler Co. Ltd.

"The one feature of the Knight motor is the fact that the oil is automatically brought up from the base to lubricate those parts of the heads which require lubrication. For a number of years we have lubricated our motors in America entirely from the base. The Daimler Co., when we came here, thought it possible we should need more lubrication in the heads. I believe that all the trouble we have ever had has been the result of this introduction of oil in the head; for the reason that if you introduce oil into the valve system independently

* *Proc. of the Institution of Automobile Engineers*, 1909–10.

of the base, and you get smoke, you think you are over-lubricating the engine and are led to believe that the whole lubrication is wrong, whereas the lower part may be dry and the smoke be coming entirely from the oil which goes into the valve mechanism itself. In the Knight motor, as I have said, the lubrication is provided so that the splash system lubricates the sleeve and also the piston from the bottom. The sleeves are so arranged that when the piston comes to the top of its exhaust stroke the inner sleeve is at the bottom of its travel, and the travel is just the proper distance to bring that portion of the sleeve which goes back over the junk ring, as we call it, down to the line which the piston has reached in the upper part of its stroke. The ring of oil which the piston leaves on the up stroke by the reciprocation of the sleeve is carried up to the junk ring above, the sleeve acting as a conveyor. The sleeve comes down, and the piston goes up; the piston brings up a ring of oil, and the sleeve comes down to get it. The piston goes down, and the sleeve carries the oil up again to the head, and so on."

The method described above shows that careful thought had been given to a somewhat special case of sliding friction. It will be observed that the bulk supply of lubricant is conveyed not in large quantities but with great regularity to the particular point where it is essential to provide it. It is also interesting and of some value to notice that a copious supply of lubricant to the head is particularly avoided, and it would appear that this rule must be rigidly observed in a rotary valve system in order to secure (1) Freedom from oiled-up sparking plugs, and (2) Low oil consumption comparable with that of a poppet-valve engine. Points which must not be ignored are that any oil used up by the rotary valve is a direct addition to that which is usually consumed in a normal poppet-valve engine, and that an increase in oil consumption, however small, is likely to furnish a line of criticism directed not on purely technical grounds against the rotary valve.

As a guide to the amount of lubricating oil which becomes objectionable in the way of fouled sparking plugs, it is worth noting that experience with supercharged engines of approximately two-litre capacity has shown that it is undesirable to permit more than two c.c. of oil per minute to enter the engine and this for a four-cylinder engine allows only one-half c.c. per cylinder.

It may be mentioned here that the scraper ring on a piston has proved to be a remarkable oil saver and a similar application of the principle has been used on rotary valves, where the problem is somewhat different and presents greater difficulties due to the presence of the port interruptions on the surface of the rotor.

KINETIC FRICTION. The prime object of all lubrication is, of

course, to reduce friction to a minimum. The result of friction is the production of HEAT, with the necessity for dissipation of the heat units by a cooling system. Most important of all, the friction is the direct cause of power-loss and, in consequence, a source of wasted energy. For this reason it is the first essential to reduce the cause of friction to the absolute minimum, in preference to removing the resulting heat units. Removal of the heat is only a method of dissipating the losses and not a means of improving the efficiency.

Without going too deeply into the technology of kinetic friction, it is at least necessary to have a clear conception of the general terms in use, before power losses can be considered. Kinetic friction is the resistance to motion which takes place when one body is moved upon another or—"that force which acts between two surfaces so as to resist their sliding on each other". If there is no pressure or load between the two surfaces—neglecting oil drag—then there is no friction. If the pressure is rated in lb. per sq. in. it will be obvious that the smaller the area under pressure the less will be the friction and the resistance to motion. The first aim should be, therefore, to reduce to a minimum the actual area under gas pressure that may be subject to sliding motion. The force of friction F varies according to the materials, the quality of the surfaces and the conditions of lubrication under which sliding occurs—and bears a certain relation to the pressure between the two bodies. This pressure is called the normal pressure N. The relation between the force of friction and normal pressure is given by the coefficient of friction usually denoted by the Greek letter μ; that is to say $\mu = F/N$. As an example, it will be apparent that a piston ring with a depth of one-eighth of an inch will be subjected, due to the pressure from the inside, to only half the load that there will be on one with a depth of one-quarter of an inch, so that the sliding force, according to the laws of friction, will in the first case be only half that in the other. In all similar devices, therefore, the total friction will be proportional to the depth of the piston ring, or in other devices proportional to the area; an important point in all self-sealing pressure constructions.

Friction With Boundary Lubrication. Table 1 indicates the great variation which exists in established coefficients of friction for lubricated and unlubricated surfaces of different pairs of materials and under different sets of running conditions. All of the coefficient values apply to conditions at moderate temperatures. The ideal hydrodynamic theory of lubrication described previously, where the mating materials do not touch each other, seldom applies in practical applications, and this accounts for the wide range of friction values given in the table. Intermediate between perfect lubrication and dry surfaces there is a condition known as boundary lubrication, where the two metals are actually touching at the

TABLE 1

COEFFICIENTS OF KINETIC FRICTION

Type of Bearing	*Materials*	*Lubrication*	*Temp.*	*Speed r.p.m.*	*Load lb. per sq. in.*	*Coefficient of Friction*	*Authority*
Journal 4″ dia. × 6″ long	Steel on gun metal	Rape oil pad lubrication	82°F.	400	364	0·010	Beauchamp Tower
Journal 4″ dia. × 6″ long	Steel on gun metal	Mineral oil oil-borne	90°F.	20	333	0·002	Beauchamp Tower
Thrust bearing disk mean rad. 6½″	Steel on gun metal	Oil-drip feed to four grooves	—	110	75	0·035	Beauchamp Tower
—	Cast iron on cast iron	Greased	—	—	—	0·08 to 0·10	Eshbach
Harland & Wolf Ltd., thrust block	Steel on Babbitt . .	Forced feed lubrication	80°F.	140	1780	0·0035 to 0·0075	C. C. Pounder
High speed bearing . .	Steel on bronze . .	Forced feed lubrication	—	—	1500	0·0014	—
Neely's type boundary film test machine (cutting wear)	Cast iron on cast iron. Hardness 220/240 2·9 carbon	Motor oil S.A.E. 30	250°F.	ft. per min. 15·4	2530	0·173 0·154 after 18 hrs.	R.W. Dayton M.I.T. Conference, June 1940
Ditto	Ditto	Ditto	Ditto	ft. per min. 156	2530	0·104 0·004 after 78 hrs.	Ditto
Ball bearings	Hard steel balls and races	Oil	—	—	—	0·0005 to 0·0015	Muzzoli and various

highest points, although there may be a film of oil between the average surface areas.

Under boundary lubrication conditions the insulating film may be broken down locally, permitting pin-point metal-to-metal contact until fresh lubricant moves in to repair the rupture. Boundary lubrication thus marks the limit at which seizure and scoring just fail to take place.

This condition is the one which is likely to exist between the sliding parts of a rotary valve, where the running temperature may be 200°C. or even higher. This fact stresses the need for the most careful choice of mating materials for the rotor and associate components and, in addition, for paying great attention to the quality of surface-smoothness. There is little data available as to the actual coefficient of friction at these temperatures, and failing more precise information the value for design purposes may be taken as lying somewhere between 0·02 and 0·06 according to the materials employed, the quality of the surface finish and the temperature prevailing.

Materials

Associated with the subject of friction is the question of the most suitable materials to be employed for minimizing the sliding forces.

A wide variety of metals has been tried and combinations for rotor and bush have been arrived at which give good resistance to wear and a life comparable with the rest of the engine.

CYLINDER HEAD. Cast aluminium for the cylinder head is particularly suitable for the rotary valve engine, for the primary reason that there are no valve seatings required as for the poppet-valve engine. The high thermal expansion of aluminium is in some respects no disadvantage, because the rotor, usually made from a ferrous material, is generally at a higher temperature than the housing, and therefore some compensation results from the differential thermal characteristics. Secondly, aluminium has a heat conductivity of between four and five times as great as that of cast iron, and therefore reduces the temperature gradient across the combustion chamber wall, keeping the inner surface cooler. Thirdly, it tends to equalize the temperature, thereby preventing hot spots.

LINERS OR BUSH. If the rotor is in direct contact with the cylinder head, as distinct from contact with a sealing device only, then it is often found necessary to include a bush or liner in the aluminium head. For this purpose leaded bronzes, phosphor-bronze tube and also white metal have all been employed with

success, but the specific material for the liner, bush or sealing device must be selected in combination with the particular material used for the rotor, to ensure the best running characteristics and least wear under the onerous conditions prevailing.

ROTORS. For the rotors, nitrided iron or nitrided steel are both excellent, giving low friction and great resistance to wear. Various special irons and steels have been manufactured with a view to obtaining a nitrided case of extreme hardness by the nitriding process. Ordinary iron or steel is unsuitable for this treatment. When iron or steel is heated in an atmosphere of ammonia there is a penetration of nitrogen into the material, but the quality of the resulting hard surface depends on various other elements which form the nitrides. Aluminium has been found by investigators
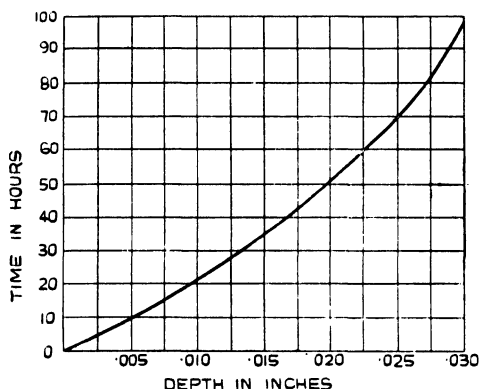
FIG. 19.
Variation in depth of case with period of nitriding.

to be the most important constituent for both stability and intensity of hardness, which its inclusion in the steel imparts to the case. Other elements in small quantities favourable to the process are molybdenum, manganese and chromium.

The actual nitriding treatment, as carried out commercially, is of a comparatively simple nature. The articles are first cleaned to remove all grease and dirt. After this preparation they are placed in a box of special material which, in addition to being heat-resisting, is also resistant to the action of ammonia gas. The cover is fixed to make a gas-tight joint. The box is then enclosed in an electric furnace which is maintained at a temperature of approximately 500°C. The box containing the parts to be nitrided is fitted with inlet and outlet tubes for the ammonia gas, which is circulated throughout the full period of treatment found necessary to give the required depth of case, as shown for a typical grade of steel in Fig. 19. The period of nitriding varies with the depth of case

required. When the period of nitriding has elapsed, the box is withdrawn from the furnace and allowed to cool down, the circulation of ammonia gas being continued until the temperature has fallen to about 100°C., when the cover may be removed.

The great advantage of nitriding over all other methods of hardening is that no quenching is required as in other processes, and the nitrided case retains its full hardness up to a temperature of 500°C., whereas ordinary case-hardened steels begin to soften at about 200°C. The surface hardness obtained is much greater than is possible with any other process, the diamond hardness number ranging from 1000 to 1100. The only movement which takes place during nitriding is a slight uniform growth of approximately 0·001 in. in diameter. As this growth can be allowed for in the dimensioning of the part, there is seldom need for straightening and grinding. The surface is best used without subsequent grinding, as the hardness of the case is greatest at the extreme surface and diminishes gradually as it merges into the original core material.

A special grade of cast iron also suitable for the nitriding process contains about one per cent of aluminium together with molybdenum and chromium and must be exposed to ammonia vapour for 65 hours or more, in order to produce a case approximately 0·015 in. deep and of a hardness of 800 Brinell.

One of the special steels is "Nitralloy", produced by Messrs. Thos. Firth & John Brown Ltd. to the following analysis:

Carbon.	0·20 to 0·026 per cent.
Silicon.	0·35 per cent.
Manganese	0·65 per cent.
Sulphur.	0·02 per cent.
Phosphorus.	0·02 per cent.
Nickel.	0·25 per cent.
Chromium.	1·4 to 1·8 per cent.
Aluminium.	0·9 to 1·3 per cent.
Molybdenum.	0·10 to 0·25 per cent.

The core of this material after nitriding, with a suitable treatment before the nitriding process, will provide a tensile strength of 35 tons per sq. in. or more. Fig. 20 shows the hardness of Firth's Nitralloy steel at varying distances from the surface. The coefficient of expansion is slightly higher than that of ordinary steel, but the expansion of the actual case is somewhat less than the core, as shown by the two curves in Fig. 21.

Die-cast gray iron, which can be obtained with a particularly fine grain, although not as good as the nitrided materials mentioned above, has been used for rotors with considerable success where a high degree of smoothness has been given to the surface before

"running-in", and when mated with a suitable material for the bush or the sealing device. Grinding-in with emery or other abrasive
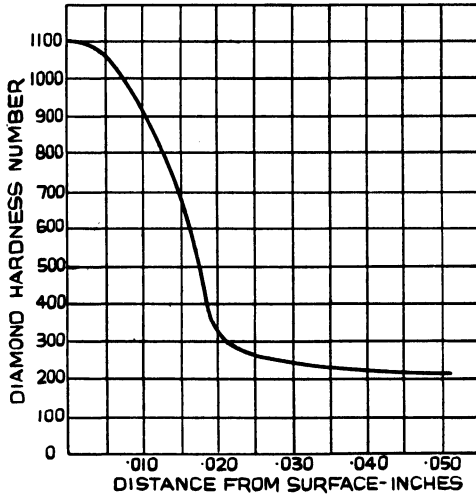
FIG. 20.
Variation of hardness with depth of case in a nitrided surface.

is not permissible in any circumstances as the lapping compound can never be entirely removed from the pores of the iron and the life of the components will in such circumstances be shortened. In
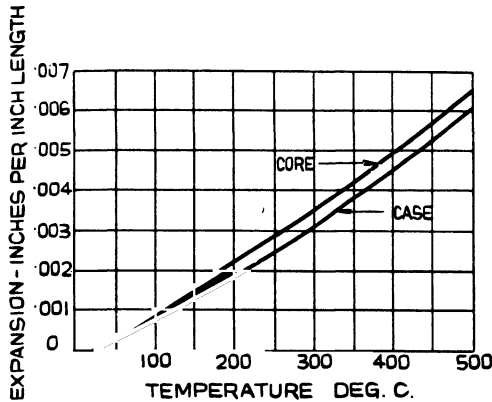
FIG. 21.
Thermal expansion of Firth's "Nitralloy" steel.

cases where die-cast gray iron is adopted in suitable combination, the requirements of rigidity, resistance to wear and low cost can be satisfactorily met.

A grade of die-cast nickel-chromium alloy iron is also available, which can be hardened and tempered to give a Brinell hardness number of 450.

Mating of Materials

Under conditions of scanty lubrication and high temperature the importance of mating the materials of the rubbing surfaces cannot be overestimated. A nitrided rotor of cast iron has been found to work well with a bush of phosphor-bronze. Experience has shown that "Carobronze", with a tensile strength of 21 tons per sq. in. and 90/190 Brinell hardness, is suitable.

When a nitrided *steel* is used for the rotor, then a softer material such as a leaded bronze is preferable for the bush and the combination has given excellent results. This pair of mating materials is probably the best that has been exhaustively tried up to the present. Time and endurance tests indicate that, with a suitable unit load, the life of the rotor and the liner is equal to, if not longer than, any other of the working components of the engine.

A nitrided rotor will also work direct on plain aluminium and give excellent life, but the latter metal has a slight tendency to scratch the hardened surface of the rotor during the running-in period. Nevertheless the combination has been employed on a number of engines with every success.

A great amount of work has been carried out to find the most suitable aluminium alloys and there is a fairly wide range from which to choose. Possibly the best is Y Alloy (B.S.,L.24), containing 3·5/4·5 per cent. copper, whilst alloys containing silicon are definitely unsuitable, as the hard silicon constituent is inclined to cut into the hardest of nitrided surfaces. There is a certain amount of difficulty in casting Y alloy if the piece is at all complicated, and in these cases the higher copper alloys are superior. The straight copper aluminium alloy, such as B.S.,L.8, containing 11/13 per cent of copper, is highly satisfactory, particularly if the design can be arranged to take care of the thermal expansion, namely 0·000023 per degree C., and this material does not lead to any exceptional difficulties in casting. It is, however, desirable to raise the casting to the approximate working temperature for several hours before the final machining operation, as this prevents permanent distortion later.

Where temperatures are not too severe, as in the case of a liquid-cooled bush, the bush may be cast in bronze and lined with white metal or Babbitt, preferably not more than 0·010 in. in thickness, and it has been found satisfactory in combination with a nitrided rotor. It is of value to note that where the design of a rotor is somewhat complicated, it may be cast in aluminium or

ordinary cast iron and shrouded by a separate shell of nitrided material.

Some experimenters have found that a chromium deposit on cast iron is definitely superior to nitrided iron both for hardness and anti-friction qualities. Also a combination of Carobronze with a "Fescolized" rotor has been tried and has given excellent results, being capable of running for periods without any oil, apparently without detriment, whereas in a comparative test with ordinary hard iron under similar dry conditions a failure occurred in a few minutes. Against this experience, a second investigator experimenting with Fescolized iron and Carobronze reported disappointing results, as in every case a series of fine scratches appeared on the surface. These contradictory reports would indicate that governing the results there are other important factors, such as surface finish, unit loading, speed and temperature, all of which have an influence on the performance of mating materials in different degree.

One firm of manufacturers, the Minerva Co., of Antwerp, has employed a case-hardened rotor of special steel working in conjunction with ordinary cylinder cast iron, and after rigorous road tests under artificial conditions of excessive road dust has proved that other parts of the engine failed due to the dust-laden atmosphere, without any defect occurring in the rotary valve gear.

Any form of sealing device which is held by pressure against the rotor is analogous to a portion of a bush or liner, and therefore the rules governing the correct mating of materials apply.

SURFACE SMOOTHNESS. It has been stressed that under the adverse conditions of boundary lubrication the materials influence the friction coefficient to a greater extent than is the case in hydrodynamical operation, when the only physical properties of the system of importance are the geometrical dimensions and the viscosity of the lubricant. But of equal or greater importance under boundary conditions is the factor of surface smoothness.

While the influence of surface finish on bearing performance has been recognized for a long time, it was not possible to make a thorough study, because a quantitative measure of surface roughness was lacking. Recently a number of methods have been developed for studying surface finish by optical or electrical amplification of the motion of a sharp point traced over the surface. In this country the roughness is usually classified and defined in average units micro-inches.* By these means it is now possible to compare one surface with another. A few examples are given in Table 2 to show the extent of the variations for different methods of production of some typical components.

* Schlesinger, G., 1942; "Surface Finish". *Report of the Research Dept. Inst. Production Engineers*, London.

TABLE 2

SURFACE FINISH OF MATERIALS PRODUCED BY DIFFERENT METHODS

All Recordings taken on T.T.H. Meter

Description of Part	*Machine Operation*	*Micro-inches (h_{ave} Units)*
Slip gauge	Lapped	0·5
Brake drum—mild steel	Ground	40·0
Swivel pin—steel	Ground	8·6
Piston—aluminium	Diamond turned	3·7
Piston—cast iron	Turned	18·0
Cylinder bore—cast iron	Turned	66·0
Poppet-valve stem	Ground	12·0
Aero cylinder bore—steel	Ground	1·5

A study of the effects of surface finish on the performance of a bearing has been made in the U.S.A. by Burwell and others.* As a result of a great amount of experimental work using bush bearings of tin-base Babbitt with surface roughness six to ten micro-inch, and steel shafts all treated to 200 Brinell, but with finishes produced in various ways ranging from 130 down to one micro-inch, they were able to arrive at the following conclusions. That the surface finish has little or no effect while the bearing is operating under hydrodynamic lubrication, but that in the lower limit of the region of hydrodynamic lubrication, i.e. boundary lubrication, for a given journal-bearing combination, the friction is markedly dependent on the SURFACE FINISH of the journal, the value decreasing as the surface becomes smoother. This implies an increase in the load capacity of the bearing with increasing smoothness and emphasizes the importance of reducing the surface roughness to less than, say, 15 micro-inches. (The micro-inch measurements referred to in the above investigation by Burwell are taken on an instrument developed by Abbott,† which gives as a reading on a meter a number which is related to the root-mean-square deviation of the surface from a median plane.)

Although the deductions are based on experiments at oil film temperatures not exceeding 249°F. (approx. 121°C.), the observations indicate that a roughness greater than 15 micro-inches would be quite unsuitable for the surface of a rotor, running at the much

* Burwell, J. T., and others, "Effects of Surface Finish". *Journal of Applied Mechanics, A.S.M.E.*, June 1941.

† *The Profilometer*, by E. J. Abbott and others. *Mechanical Engineering*, vol. 60, 1938, pp. 205–216.

higher temperature and under the more onerous conditions which prevail in the head of a combustion engine.

Power Losses

Proceeding next to a consideration of the power losses in driving a rotary valve, it has been shown that the friction coefficient is a variable quantity, depending on many factors and conditions. It is, in consequence, seldom possible to make accurate quantitative analyses of work losses, but by adhering rigidly to the fundamental rules of friction it is practicable to design for optimum results and also to calculate the comparative power losses inherent in one design or another, assuming always some nominal coefficient of friction.

Various grades of cast iron have frequently been used in combination for the rotor and the stator of rotary valves in the past, and for the purpose of evaluating the power losses by way of a typical example the friction coefficient of cast iron on cast iron under boundary conditions of lubrication may be taken as μ=0·050. The problem will be based on the application of an elementary flat disk valve of the self-sealing type illustrated in Fig. 4.

DATA. Single cylinder engine. Four cycle.
Area of rotary disk, 12 sq. in.
Average pressure of power stroke, 90 lb. per sq. in.
Mean circumference of rotation, 7 in.
Crank shaft speed of engine, 3000 r.p.m.
Speed of rotor = 1500 r.p.m.

Find the work done in turning the disk through one revolution and the power loss per minute.

"Work" in mechanics is the product of force and distance and is expressed in foot-pound units. In this case:

$$7/12 \times 90 \times 12 \times 0{\cdot}050 = 32 \text{ ft. lb. approx.}$$

This quantity must be divided by four, as the average explosion pressure is only acting on the disk during the firing stroke; that is, during one quarter of a revolution of the rotor. The work done in turning the valve during the compression stroke can be assessed in the same manner taking the average pressure during compression. Say that this proves to be 3 ft. lb. The total work done per revolution of the valve will be 32/4 ft. lb. + 3 ft. lb. (i.e. 11 ft. lb.) The power loss at a valve speed of 1500 r.p.m. will then be 11 × 1500 = 16,500 ft. lb. per minute, or approximately 0·50

of a horse-power. In this calculation the reduction in area of the disk, due to the gas aperture, has been neglected. The power loss with this design of valve is very high and there is little that can be done to minimize it, for the fundamental reason that if the diameter of the disk is reduced appreciably, port area is sacrificed, and therefore the design is inherently bad. From a consideration of the calculated results shown in the above example it will be appreciated that the factors governing the power loss from a rotary valve are:

(1) The coefficient of friction.
(2) The net area of the sealing member, subjected to the fluid pressure.
(3) The surface speed of the contacting surfaces.

The total power loss is directly proportional to each of the above factors and from a theoretical point of view these should all be kept down to the absolute minimum.

The power losses of other types of rotary valves may be computed by methods similar to those employed in working out the above example.

In order to afford some idea of the power losses in the valve gear on a poppet-valve engine and the performance to be aimed at when employing a rotary valve, it is well to consider the work of Ricardo on a 100 horse-power, six-cylinder engine. The losses in respect of the various parts of the engine due to friction, expressed in lb. per sq. in. mean pressure, are shown in Table 3.

TABLE 3

LOSS DUE TO FRICTION FOR A 100 H.P. SIX-CYLINDER ENGINE (RICARDO)

Bearing friction	0·75 to 1·00	lb. per sq. in. mean pressure
Valve gear	0·75 ,, 0·80	,, ,, ,, ,, ,, ,,
Magneto	0·05 ,, 0·10	,, ,, ,, ,, ,, ,,
Oil pump	0·15 ,, 0·25	,, ,, ,, ,, ,, ,,
Water pump	0·30 ,, 0·50	,, ,, ,, ,, ,, ,,
Total	2·00 to 2·65	,, ,, ,, ,, ,, ,,

It will be observed that the valve gear absorbs from 0·75 to 0·80 lb. per sq. in. mean pressure and when expressed as a proportion of a b.m.e.p. of, say, 120 lb. per sq. in., gives a friction loss of considerably less than one per cent of the horse-power of the engine. If due attention is paid to the main factors affecting the power loss with a rotary valve, then it is possible to approach the level of efficiency attained by the poppet-valve engine.

(III) *EXPANSION AND DISTORTION DUE TO THERMAL EFFECTS*

The temperature of combustion is very high, possibly 2000°C., and poppet-type exhaust valves are known to reach a temperature of 700°C. at the centre of the head, where it is impossible to provide direct liquid cooling. Similarly the centre of a cast-iron piston crown may reach a temperature of 500°C. under full load conditions.

Nearly all metals expand more or less with increase of temperature and the coefficient of thermal expansion per degree C. is used to express the linear change in size. This implies a proportional increase in diameter or length for each degree increase in temperature. The coefficient of thermal expansion is usually taken as an average over a certain range of temperature.

The order of magnitude of thermal expansion for some typical materials in general use is shown in Table 4.

TABLE 4

LINEAR COEFFICIENT OF THERMAL EXPANSION

In per in. per degree C.

Material	*Coefficient*	*Temperature Interval Between*
Aluminium	0·0000257	20°C. and 300°C.
Brass 65/35	0·0000188	20°C. and 100°C.
Phosphor bronze	0·0000168	20°C. and 100°C.
Cast iron (ordinary)	0·0000116	25°C. and 300°C.
Steel	0·0000126	20°C. and 100°C.
Nitralloy steel	0·0000130	20°C. and 300°C.
Aluminium 5% sil. (cast)	0·0000219	20°C. and 100°C.
Aluminium 5% sil. (cast)	0·0000228	20°C. and 200°C.
Aluminium 5% sil. (cast)	0·0000238	20°C. and 300°C.
Nickel-chrome-iron alloy	0·0000190	20°C. and 300°C.

(Coefficients per degree F. are 5/9ths of those per degree C.)

It will be seen that ordinary cast iron is a good material to use where increase in size and possible distortion at high temperatures must be kept low. In cases where aluminium is used instead for the piston of an I.C. engine it is known that the diametral working clearance must be increased approximately in the same proportion as the coefficients of expansion of the two materials. The magnitude of the clearance, in the case of a piston, is relatively unimportant from a pressure leakage point of view, because the piston ring should effectively seal the clearance at all temperatures.

Just as it is necessary to provide a clearance sufficient to cover the expansion and distortion of a piston working in the cylinder of a steam or I.C. engine, it is of equal importance to arrange for a similar clearance in connection with a cylindrical rotor revolving in a cylinder head. The temperature range to be covered may be substantially the same in both cases, say from cold at 10°C. to 310°C., which amounts to a temperature rise of 300°C. In order to afford some idea of the clearance required for a piston head of cast iron three inches in diameter, let us take the tabulated figure for the thermal expansion of cast iron as 0.000011. The expansion will then be $3 \times 0.000011 \times 300 = 0.010$ in. To this must be added, say, 2/1000 of an inch for probable distortion due to varying temperature at different points, and another 1/1000 in. for oil clearance at maximum temperature and to cover manufacturing tolerances. We thus arrive at a necessary total diametral clearance of 0.013 in.

From the above it will be evident that any attempt to make a rotary valve of cylindrical form pressure tight by superfine workmanship and without the addition of an effective sealing device round the ports is a futile proposition. The failure of many designs can be ascribed to the omission of proper consideration of this important point.

There are, of course, special materials available at the present time with a thermal expansion less than that of nitrided steels and cast iron, which will to some extent reduce the foregoing calculated clearance of 0.013 in. In addition, adequate liquid-cooling can be expected to reduce the previously suggested temperature range of 300°C. to something nearer 200°C. However, quite a small clearance between two surfaces offers a surprisingly large space through which gas may escape. For example, take a typical conical valve with a circumference of nine inches at the base of the cone and with a clearance of 0.002 in., which is approximately the clearance allowed on a big end bearing of an automobile engine. Then, the space for leakage would be equal in area to a drilled hole 1/8 in. in diameter. Should a blowhole of this size develop in the head of a piston the results would hardly be considered satisfactory. This only serves to demonstrate the importance of proper sealing.

It may be here mentioned that nickel-iron alloys containing from 35 to 36 per cent nickel possess a minimum coefficient of expansion of approximately 0.000002 per degree C., as shown by Guillaume* (see Fig. 22). These special materials, however, are very critical, and the presence of impurities such as manganese and carbon make it difficult to obtain the minimim expansibility, and a coefficient of expansion of 0.000006 per degree C. is a more reasonable figure to use for design purposes. It should also be emphasized

* Vide *Proc. Phys. Soc.*, London, 1920. C. E. Guillaume.

that these alloys have such coefficients of expansion only over certain temperature ranges, some of them only up to 200°C., and after that the alloy expands comparatively rapidly. Similarly, below room temperatures, at a certain point they start to contract at a more rapid rate. It will also be appreciated that these materials are somewhat expensive and a full knowledge of their limitations is required before they can be used with success.
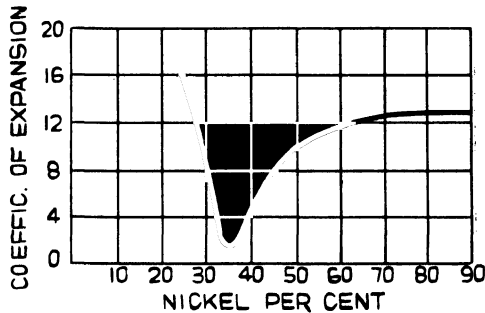
FIG. 22.
Variation with nickel content of coefficient of expansion at 20°C. in millionths of an inch. (Guillaume.)

Research has provided a special nickel-chrome-iron alloy with a coefficient of expansion of 0.000019 per degree C., which is particularly suitable for sleeve valves, as this is only a little less than that of the aluminium alloy used for the cylinders of aero engines. The steel used for the sleeves of the Bristol Hercules contains 11·5 to 13·5 nickel, 4·5 to 5·5 chromium, 0·25 to 0·40 vanadium, 0·4 to 0·55 carbon and 5 to 6 per cent manganese. With this material it has been found possible to run with a clearance on diameter between sleeve and cylinder bore of from 0·004 to 0·006 in.

CHAPTER III

PORT AREAS AND VALVE DIAMETERS FOR FOUR-CYCLE ENGINES

WHEN considering the subject of port areas, it is necessary to bear in mind the physical shape of the combustion chamber of an engine and the compactness in form now found essential to meet modern compression ratios of the order of seven to one and higher. This requirement allows somewhat limited space for the sparking plug and the valve openings, and it may, under some conditions, lead to a stage where the inlet and discharge areas of valves or ports have to be constricted in order to avoid increasing the volume of the combustion chamber.

Maximum Valve Area

The maximum valve area that it is practicable to accommodate in a given diameter of cylinder depends to some extent upon the physical arrangement, the design and the position of the valves, whether of poppet or rotary type. A ready method for making a comparison of alternative arrangements is to evaluate the relative area of the valve to the area of the engine piston. The quotient which results from dividing the valve area by the piston area is then known as the valve/piston area ratio. Take as an example a racing type of engine with conventional overhead poppet-valves as shown at A in Fig., 23 where provision is made for one inlet and one exhaust valve per cylinder, the valves being inclined at an included angle of, say, 100°. Assume the bore of the cylinder to be 65 mm. and the throat diameter of the valves 36 mm. Let a equal the gross area of the poppet-valve aperture and A equal the area of the piston, then: $a/A = (36 \times 36 \times 0{\cdot}785) / (65 \times 65 \times 0{\cdot}785)$, giving a ratio of 0·31. This value of 0·31 is somewhere near the maximum which it is possible to obtain with any type of valve in a high-compression engine and it approaches the optimum for all diameters of pistons. In engines of very large bore using a standard size of sparking plug it may be possible to gain some slight relative increase in the area of the valve due to the fact that the space required to accommodate a standard sparking plug is a constant for all diameters of piston.

It should be mentioned that there are many engines in general use giving excellent service in which the a/A ratio of both inlet and exhaust is as low as 0.20, in cases where maximum breathing efficiency has not been particularly sought. On first consideration

it may be suggested that in the design of a rotary valve the aim should be to provide the maximum valve/piston area ratio mentioned above in order to compete with existing high efficiency poppet-valve engines. For a valve/piston area ratio of 0·31 in an engine of 65 mm. bore, a rectangular port of dimensions 50 mm. long by 20 mm. chordal width will be necessary as indicated at B in Fig. 23. Although an aperture of this size is not impractical, it is by no means an easy matter to accommodate a port of such dimensions when the problem of sealing is to be overcome with a minimum of friction.

There is, however, ample evidence (unfortunately not on a
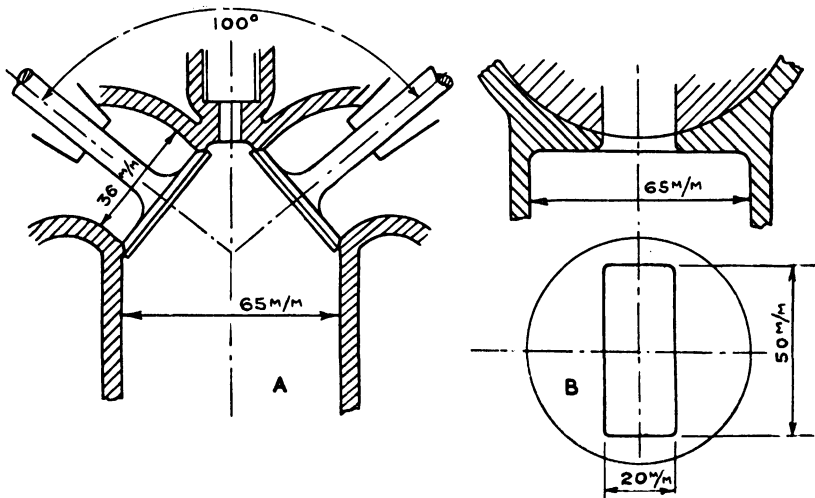
FIG. 23.
Comparative dimensions of poppet-valve and port of rotary valve with valve/piston area ratios of 0.31.

rigid quantitative basis) that the area of an open port in a rotary valve is of greater value for the flow of the gases than the equivalent effective area of a poppet-valve, and that in the design of a rotary valve there is no need to conform to the a/A ratio found advantageous with the orthodox lift valve. As an example it is claimed that in the Bristol air-cooled engine Hercules Mark XVII the single sleeve valve has a better gas inlet and a better discharge coefficient than the four poppet-valves in earlier engines of the same make. Professor W. Morgan has given a great deal of time to the study of valve efficiency in connection with sleeve valves, and he has made comparisons with data from German sources for Adler and Benz poppet-valve engines, from which it is deduced that one square cm. of port area of a sleeve valve is worth two square cm. of port area of a poppet-valve, and that in consequence, with the former,

the filling of the cylinder is possible at much greater speeds. Other investigators have shown that the port of a rotary valve has a similarly improved breathing characteristic.

COEFFICIENT OF EFFLUX. In calculating the gas flowing in feet per second for small differences of pressure upon two sides of an orifice or pipe, a factor C, known as the "coefficient of efflux", is introduced into the usual formulae. The coefficient of efflux, C, varies with the type and dimensions of the port. In general, the poppet-valve port is obstructed by the valve and the guide, whereas the port of the rotary valve is unobstructed. An important difference exists between the efflux capacities of such ports. Weisbach has given various values for C as shown in Table 5. The annular space

TABLE 5

COEFFICIENT OF EFFLUX (WEISBACH) FOR SMALL DIFFERENCES OF PRESSURE

Shape	*Value of C*
Circular orifice in thin plate	·56 to ·79
Short cylindrical pipe	·81 to ·84
Short cylindrical pipe rounded at exits	·92 to ·93
Conical converging mouthpiece	·90 to ·99
Streamline short pipe to shape of contracted vein	·97 to ·99

between the periphery of an open poppet-valve and its seating is likely to provide a coefficient of efflux considerably less than that for a circular orifice in a plate, which is given in the table as from 0·56 to 0·79, whereas the open port of the rotary valve is likely to approximate the value of 0·81 to 0·84 given for a short cylindrical pipe. By careful streamlining, the latter figure might be further improved.

From the above data it may be estimated that, other things being equal, the weight of charge passed by a rotary valve compared to a lift valve of similar area will be between twenty and twenty-five per cent greater.

DIAGRAM OF VALVE MOVEMENT. The rate of opening of a valve is also a factor influencing the weight of charge passed into the cylinder. There is a subtle difference between the progressive change in area produced by a cam movement for a lift valve and the sliding motion of a rotary valve. The profile of a cam has to be formed to give a slow rate of opening and a slow rate of closing to ensure quietness of operation. If the cam is made with an abrupt rise, the acceleration of the moving parts may become

so great that, owing to the momentum, they actually leave the surface of the cam. A typical curve showing the varying area resulting from the movement of a cam-operated poppet-valve is illustrated in Fig. 24. The ordinates represent the area of the aperture at different points in the angular travel of the camshaft
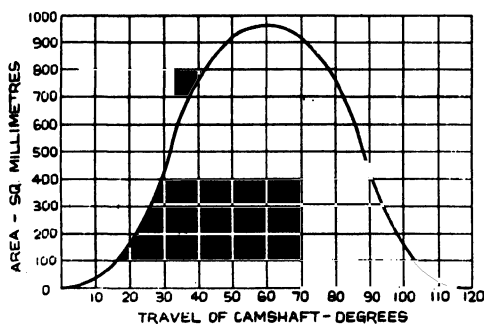
FIG. 24.
Variation in valve area of poppet-valve with angular travel of camshaft.

in degrees. It will be seen that the area of discharge at the commencement and at the end of the period of operation increases or decreases at a very slow rate. The action of the port in a rotor has quite a different characteristic. The increasing and diminishing area at opening and closing of the port is shown in Fig. 25. It will
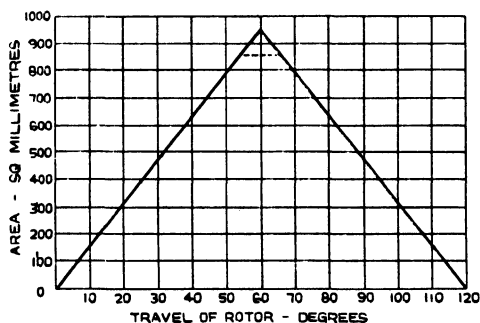
FIG. 25.
Variation in port area with angular travel of rotary valve.

be seen that the graph here follows a straight line law, with a comparatively rapid and a uniform increase in area from the start up to full aperture and a similarly rapid rate during the closing period. It should be emphasized that the period at full aperture for the rotary valve is only instantaneous, except in circumstances when the dimensions of the moving and stationary

ports are of dissimilar chordal width. It is then feasible to provide a pause at full aperture, but, since this modified characteristic is gained only at the sacrifice of optimum area, as indicated by the dotted line at the apex of the graph in Fig. 25, the feature is of negative value except as a means of establishing a difference in angular timing for exhaust and inlet.

Sliding Velocity

In determining the design of a rotary valve movement it is first necessary to decide on a chordal width of port to suit the desired area of exhaust or intake. It is of importance to note that a long port with minimum chord provides the lowest surface speed for a given area. When once the chordal width has been decided upon, then the velocity between the sliding surfaces for a given engine speed is established regardless of the diameter of the rotor. This requires some further explanation. Actually there is, according to one or other selective port arrangements in the rotor, a choice of two diameters, one diameter being almost twice that of the other, as shown in Fig. 5. If the port-arrangement A is adopted—that is, the one with the rotor of largest diameter—then the speed of rotation will be quarter crankshaft speed, whereas if the scheme shown at B is selected, then the speed of rotation for the rotor will be half crankshaft speed; consequently in both cases the surface velocity relative to the speed of the engine is the same, or nearly so.

In practical designs there are a number of variations on these arrangements, but the two basic examples A and B illustrate the comparative diameters for a given chordal width of port, with a consequential change in the driving gear ratio. It is sometimes claimed that the quarter-reduction system allows a reduced surface velocity for the rotor, but that is an erroneous deduction which cannot be substantiated. The sliding velocity, as stated, is in both types almost, but not exactly, the same for a given chordal width of port. The very small difference which exists will be explained later.

In general, rotors of form A have the advantage that they provide more space for purposes of internal liquid cooling, and with this form it is also possible to provide a better streamline path for the flow of the inlet and exhaust gases than is possible ith rotors of form B, Fig. 5.

Timing of Rotary Valves

In making a study of timing conditions it is necessary to bear in mind that in the basic arrangement A, Fig. 5, the ports pass

through on a diameter and in consequence there are two entries or exits to the cylinder per revolution of the rotor; hence the specific requirement of a gear ratio of quarter-engine speed.

In Fig. 5 B the port channel enters the rotor and then is diverted through a right angle before making an exit. The inlet and exhaust channels in these circumstances are separated in the rotor by a longitudinal or diagonal barrier. The branch ducts for induction and exhaust are spaced apart on the valve chest to agree with the staggering of the ports in the rotor. An alternative to scheme B, which has been used extensively by one constructor in conditions when the rotor is designed to function for a single cylinder, is to arrange for the live charge to enter at one end of the rotor and for the exhaust to leave by the other end. In these circumstances the driving-spindle is usually arranged to pass through the inlet duct as shown at D, Fig. 5. In either of these types B or D the layout of the internal duct in the rotating member provides one entry only to the cylinder per revolution of the rotor, the speed of rotation being, therefore, half that of the crankshaft and in this respect similar to a camshaft in the ordinary poppet-valve engine. Although drive ratios of two-to-one and four-to-one are common for valve operating speeds, a rotor can be designed to rotate at any even figure ratio of reduction from crankshaft speed. The higher the ratio the greater the diameter, therefore a ratio of six-to-one, for which provision has to be made for three entries per revolution of the rotor, is probably the largest which has been used up to the present, the diameter of the rotor varying directly with the number of entries.

There is one other basic type of rotor which demands consideration, and this is shown diagrammatically in Fig. 5 C. All of the rotors previously described have a notable similarity in that the duct in the rotor for induction is quite separate from the internal duct which carries away the exhaust gases. In consequence, there is no intermingling of the live charge with the products of combustion, except in so far as some leakage may exist between one and the other if the sealing is imperfect. In the form of valve C the pocket formed in the rotor is used alternatively for the transfer of the charge from the carburettor to the cylinder and later for the transfer of the exhaust gases from the cylinder to the atmosphere.

This introduces a new disadvantage, and it means that the volume of the pocket is wastefully emptied of live mixture at every exhaust stroke and likewise at each induction stroke a similar volume of exhaust gases is drawn into the cylinder. This loss of charge may not be great but it is a fundamental disadvantage of some significance.

Timing Specification. The timing specification for a rotary valve differs but little from that used for a poppet-valve engine.

In general, a valve setting that is satisfactory for a poppet-valve engine at 2000/3000 r.p.m. can be applied to a rotary valve engine running at much greater speeds, say up to 4000/5000 r.p.m., but valve timing diagrams may vary considerably with different port arrangements and different types of engines. A typical specification which has been used for speeds of 4000 r.p.m. and over is given by way of example: Inlet valve opens 12° early and closes 48° late. Exhaust opens 48° early and closes 12° late. It will be noted that the full period for both the inlet and the exhaust is the same, namely 240° on the crankshaft, and this is not unusual practice, enabling all of the ports to be made of similar chordal width, and therefore of the largest possible area at full aperture. It is probable that if a rotary valve were designed with much smaller port area than is customary, one or all of the following modifications—considerable overlap, earlier opening of the exhaust and later closing of the inlet—would produce the most favourable performance characteristics. The timing specification would in fact be more in line with poppet-valve practice for high-speed running conditions.

Diameter of Rotor. It is important to be able to calculate the precise diameter of a cylindrical rotor for a given chordal width of port and to meet a given timing specification for either a two-to-one or the four-to-one system. Let the following data be taken for two examples:

Data.		
	Chordal width of port	20 mm.
	Movement of crank for full period of valve opening	240°
	Driving ratio ..	4 to 1 with double-entry rotor.
	Driving ratio ..	2 to 1 with single-entry rotor.

The corresponding angular movement of the double-entry rotor will be 240/4 = 60° and in the single-entry rotor 240/2 = 120°. It will be realized that the angle embraced by the port will in each case be only half the above, due to the width of the stationary port, adding to the period during which the port remains open. Referring to the diagram Fig. 26, the geometry for both of the alternatives is shown. In case A the included angle will be 60/2 = 30° and in case B the included angle will be 120/2 = 60°, i.e. twice that for case A.

Let θ = half the included angle in degrees.
C = chordal width of port in mm.
R = the radius of rotor in mm.

Then, for case A:

$$R = C/(2 \times \sin. \theta) = 20/(2 \times 0{\cdot}258) = 38{\cdot}64 \text{ mm.}$$

For case B:

$$r = C/(2 \times \sin. \theta) = 20/(2 \times 0{\cdot}500) = 20 \text{ mm.}$$

It will therefore be seen that the diameter of the rotor A is nearly, but not quite, twice that of rotor B, the diameter varying inversely as the sine of half the angle embracing the chord.

The power loss due to friction, other things being equal, will be for all practical purposes the same, regardless of whether type A or type B rotor is adopted, because the surface speeds are approximately equal.

It has been mentioned previously that some slight variation from equal width of port for rotor and stator will be necessary, when the period of opening of inlet and exhaust are not identical, but this is a small matter presenting little difficulty, and any normal
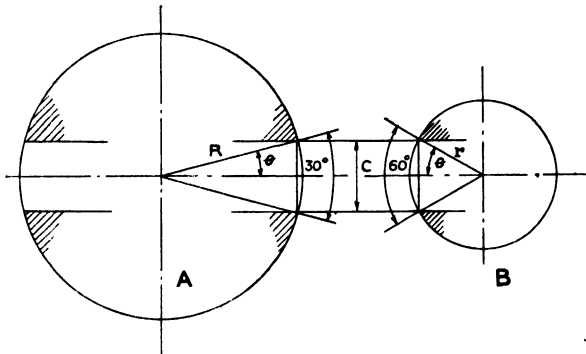
FIG. 26.
Evaluation of diameter for a given chordal width of port.
A—Double entry rotor. Drive ratio 4 to 1.
B—Single entry rotor. Drive ratio 2 to 1.

valve timing specification can in practice be provided for, with small departures from equal sized ports.

Balancing of Rotors

The correct balance of a rotary valve, from both static and dynamic viewpoints, is just as important a matter as the correct balance of a crankshaft. It is to be appreciated that it is impossible to manufacture dynamically perfect rotors because:

(1) The material is never quite homogeneous.

(2) There are always geometrical errors.

(3) The rotor may distort under operating conditions. It must, however, be borne in mind that balance weights as small as 1/50,000 of the rotor weight* can have noticeable effect.

A rotary valve is in static balance if it is so proportioned that

* Kroon, R. P., Dec. 1943. "Balancing of Rotating Apparatus". *Journal of Applied Mechanics Trans., A.S.M.E.*

if it is placed with its axis horizontal on a pair of steel knife-edges it will remain in any position in which it is placed. By the addition of suitable counterweights it is not a difficult matter with most forms of rotary valves to obtain static balance, but to ensure dynamic or rotating balance is a more difficult proposition.

The general accepted principle of rotating balance may be stated as follows:

If a thin section is cut from the rotating body, perpendicular to its axis of rotation and at any point along its length, and then any line is drawn across the section and through the axis, the sum of the products of all mass particles of the section to one side of this line and their respective distances from the line (their moments) must be equal to the sum of the products of all particles on the other side of the line and their respective distances from the line.

It is obvious from the above that the only form of rotor which fully satisfies this condition in a practical manner is one in which each slice across the axis is symmetrical.

The rotor shown in Fig. 5 A is basically correct in form and is actually nearer to perfection than the orthodox balanced four-throw crankshaft. Referring to Fig. 5, rotors of the forms B, C and D cannot theoretically be made to give perfect dynamic balance, as it is impracticable to place a mass to compensate for the open space of the port, but due to the comparatively great rigidity of a cylindrical rotor it is possible to obtain nearly the same result by placing weights on either side of the port. However carefully the balancing of one section against another may be done on paper, it is most unlikely that the actual manufactured component will be as accurate as the drawing, especially as the inside contour can in general practice be produced only by casting and not by machining. At prospective engine speeds of 6000 r.p.m. quite a small unbalanced component can produce severe vibration, and if this were permitted to exist, one of the chief merits accruing from rotary motion would be nullified, at least for the higher engine speeds.

With respect to disk and cone rotors with a single port as shown in Fig. 4 it is not easy to balance such parts statically. It is quite impracticable to design for perfect dynamic balance, although with care a near approximation has been gained by the utilization of inserts of relatively greater specific gravity than the material from which the rotor is made, these inserts being located at each side of the slot.

CHAPTER IV

DRIVE GEAR AND GENERAL APPLICATION OF THE ROTARY VALVE TO THE ENGINE

THE application of the rotary valve to the I.C. engine requires the consideration of a variety of drives, since rotors are of diverse form, may run at one of several velocity ratios, may be arranged with their axes vertical or horizontal, and may, if horizontal, be situated at right angles or alternatively in line with the crankshaft.

Drive Arrangements

A selected type of valve may be eminently suitable for practical application to an engine with a single cylinder, but may present great difficulties and complication in the design of the driving gear when applied to a multi-cylinder engine of the all-in-line type. Again, it may offer still greater mechanical problems when the adaptation is made to an engine of the class in which a number of cylinders radiate from the axis of the crankshaft. In these circumstances it is not possible, nor is it advisable, to say that one form of rotary valve is preferable to another without a close detailed study of the type of engine to which the valve is to become a part. Broad observations can, however, be made.

ALL-IN-LINE ENGINES. It is logical to suggest that for a multi-cylinder power unit of the all-in-line or banks of cylinders-in-line type of engine the horizontal cylindrical rotor located above the combustion head and running the full length of the block, parallel to the axis of the crankshaft, is the most suitable form. This is also the type of valve which calls for the most elementary driving gear. The drive may consist of a simple train of plain spur gear-wheels or even a single driving chain; or, again, it may be a combination of spur gears with a chain, operating between the crankshaft and the rotor. Any one of these combinations affords a system with few bearings, devoid of bevel or spiral gears and not necessitating a vertical shaft. In general, any type of established drive, as at present employed for an overhead camshaft on a conventional poppet-valve engine, can be used for a cylindrical rotor with good effect.

The absence of any need for adjusting independently the valve setting for each of the cylinders in one line or bank is a noteworthy feature in assembly and servicing, and is a point which will appeal strongly to the maintenance staff in charge of aero engines which

now usually comprise a great number of cylinders and a greater number of valves, each requiring individual adjustment.

DRIVES FOR HORIZONTAL ROTORS. Several representative drives suitable for a horizontal rotor of cylindrical form are shown diagrammatically in Fig. 27. When chains are employed, the arrangements are readily adaptable for velocity ratios of either two-to-one or four-to-one without any important structural alterations to the engine. The single chain drive shown at (a) is the
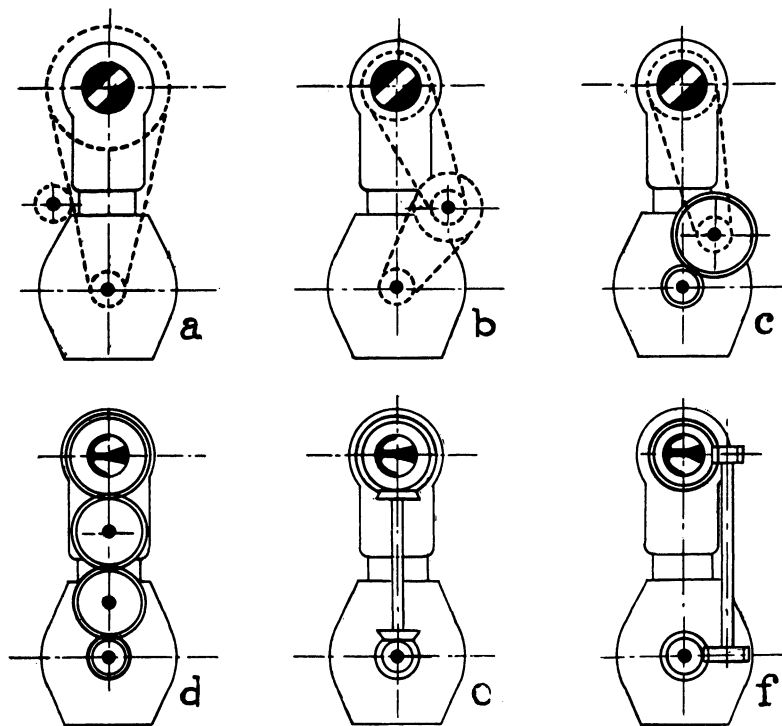
FIG. 27.
Typical drives for cylindrical rotors. (a) Single chain and jockey pulley. (b) Two stage chain drive. (c) Spur gear drive and chain. (d) Train of spur gears. (e) Double bevel drive. (f) Double skew gear drive.

most elementary, but it has one disadvantage: that on a long-stroke engine the chain becomes unduly long and a jockey-wheel for adjustment of the chain tension is almost a necessary addition to the scheme. Short chains are always to be preferred, and the two-stage chain drive indicated at (b) is a system of considerable merit, allowing easy removal of the top chain in the event of any need to dismantle either the cylinder head or the rotary valve. The lower chain is then housed under a separate

cover, and neither the lower chain nor its cover need to be disturbed when dismantling the head. When a system of this kind is employed, the adjustment to the tension of both chains can be readily carried out by simple means, as, for instance, by the provision in the design for a nominal lateral displacement of the spindle upon which the twin chain wheel revolves. In the two-stage system shown at (c), wherein there is indicated a pair of spur gear wheels for the lower drive, it is not so easy to make provision for adjustment of the chain on the upper drive, but it is important that this feature is not neglected. If the valve head is detachable from the cylinder block, as is now usual, the varying thickness of the copper gasket (consequent on different degrees of tightening) definitely demands some device for adjusting the tension of the chain. This requirement is probably best met by a jockey pulley. The arrangement shown at (d) is an all spur-gear layout and suffers a similar handicap, in so far as a variation in the thickness of the gasket results in a change to the depth of mesh of the top pair of spur gears. This calls for either a special form of deep tooth-profile or an inordinate amount of backlash between the teeth until the gasket has become fully compressed. However, backlash between gears is not nearly so important in drives for rotary valves as it is for camshaft drives, because the torque is more nearly constant.

Redesign from Poppets to Rotors. If a normal poppet-valve engine is to be redesigned to receive a cylindrical rotary valve, it is worthy of note that if the original engine incorporates a single overhead camshaft, then the standard engine will require few changes to the existing cylinder casting and other units. Should the head be detachable, it is generally a straightforward proposition to accomplish the redesign without resort to any but the smallest of changes to the main cylinder block or in the layout of the driving gear. In fact, in some instances it may be a practical undertaking to arrange for the design of the rotary head to be fully interchangeable with the poppet-valve unit and thereby enable much experimental work to proceed in parallel with the production of the standard engine, including the gain of comparative data, with little dislocation in production. Modifications in the assembly line will then be few when a change-over is eventually approved.

The drives indicated at (e) and (f) are typical of common arrangements which have been established for overhead-valve engines in the past and both drives are suitable for use with a cylindrical rotary valve. Allowance for variations in the thickness of the gasket between the head and the cylinder block is usually catered for in arrangements following (e) and (f) by including a telescopic joint somewhere in the vertical shaft.

The foregoing discussion refers to a horizontal valve, and it

must be realized that, however attractive and efficient the alternative vertical type of valve may be, its application to a more or less standard engine is not a practical proposition, and the manufacturer who wishes to adopt a vertical valve must be prepared to face a very extensive redesign of the whole power unit.
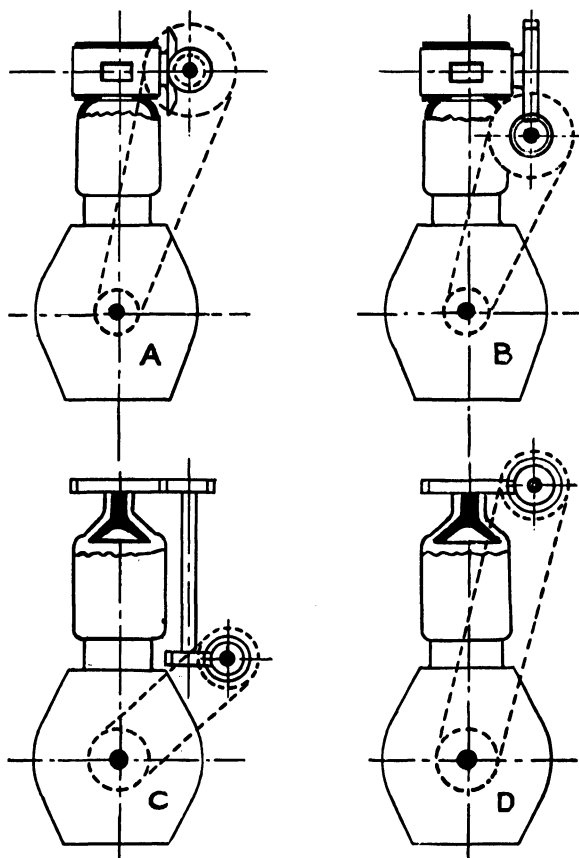
Fig. 28.

Typical drives for individual rotor per cylinder—single or multi-cylinder engines. (A) Horizontal rotor at right angles to crankshaft with chain and bevel drive. (B) Horizontal rotor at right angles to crankshaft with chain and skew gear drive. (C) Vertical rotor with chain, skew gear and spur gear drive. (D) Vertical rotor with chain and skew gear drive.

Independent Rotor per Cylinder. The independent valve placed in the head of each cylinder has strong advocates for valid reasons, especially in designs for air-cooled engines, where advantages are to be derived from symmetrical cooling of each unit with the least possible thermal distortion. In particular reference to

single-cylinder engines, a rotary valve of any type possesses the inherent feature that, in all applications, the gears need to be located but once to provide the correct timing for all time, but it is of considerable importance to note that when the vertical valve of disk or conical form is employed, as distinct from the horizontal rotor, a right-angled gear drive cannot be avoided, and in addition a pair of spur gears is generally required for each cylinder. It is sometimes possible to use the vertical shaft in a dual capacity, i.e. for driving other auxiliaries, such as an oil pump, magneto, dynamo and the like. The advantages accruing from this artifice may tip the scales of general economy in the design of the engine and may favour the adoption of one type of rotor in preference to another.

In the single cylinder engine with a horizontal rotor the right-angle drive can only be avoided when the axis of the rotor is parallel to the crankshaft. When located across the cylinder head as shown in Fig. 28 A and B, bevel gears, skew gears or their equivalent are a *sine qua non.* Either of these arrangements A or B may also be used on a multi-cylinder engine with the inherent benefits of uniform cooling for each cylinder head, but with the multiplicity of driving gears, a disadvantage generally only associated with the disk or conical valve situated vertically in the head. Typical drives for the vertical type of valve are represented in Fig. 28 C and D, and both arrangements are somewhat elaborate.

In spite of the complications associated with the drive, there is much to be said for the independent rotor per cylinder, since each valve may then be withdrawn and examined separately, whereas the longitudinal horizontal rotor cannot conveniently be removed from an engine installed, say, in an automobile with the dashboard at the rear end of the rotor and the conventional cooling radiator at the front. Under these conditions the whole combustion head will need to be taken down before the rotor can be withdrawn, unless the rotor head is split horizontally, as has been done by more than one designer.

The transversely located cylindrical rotor therefore has merit where frequent inspection of independent valves is necessary, as in an engine used for racing purposes.

DUAL ROTOR. There is a rather special, but nevertheless interesting, case where a common vertical rotor is disposed between a pair of cylinders and serves each in turn. This design has been inaugurated by the Itala Fabrica di Automobili, of Turin, and has also been employed in the Russel engine of American origin. Both of these engines are described in detail in a later chapter. A four-cylinder engine requires but two dual valves and a six-cylinder engine three dual valves. The drive indicated in Fig. 28 C is generally applicable, but the number of vertical shafts and gear wheels is halved.

An arrangement of dual valve which, as far as the author is aware, has not been suggested before is shown diagramatically in Fig. 29. In this proposal a valve of cylindrical form with dual ports, as used by Itala, is situated above the cylinder head with its axis horizontal and at right angles to the crankshaft between a pair of cylinders, each valve serving two cylinders and functioning in exactly the same manner as described later in the references to the Itala and Russel engines. A good reason for employing such an arrangement is that the power losses are reduced by one half, as the dual valve does not demand a greater diameter by virtue of its serving two cylinders. The system permits the use
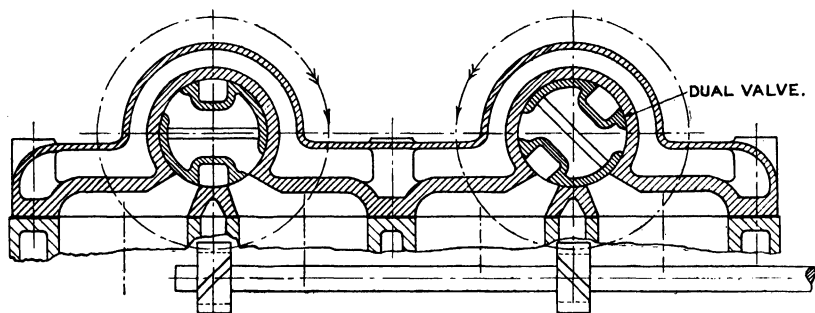
FIG. 29.
Drive for two dual valves feeding four cylinders.

of simplified intake and exhaust manifolds, reducing the number of branches by half.

Stress in Drive Gears

Each of the schemes enumerated above provides certain advantages in some directions and disadvantages in others, and there is insufficient data at present to enable a pronouncement to be made that any one system will to any extent become standardized for some considerable time. However, whatever the character of drive, it can be said that the technique necessary to design a silent arrangement is certainly less difficult than is called for when designing a camshaft and its associated tappet gear, where a periodic reversal of torque is an invariable accompaniment. The torque of almost constant value without any characteristic flutter permits the gears and shafts for a rotary valve to be designed on a strictly dead stress basis without the usual allowance for impact loads. This permits smaller gear teeth and less width of face.

In considering the design of a drive and mounting for a rotary valve there is a technical point which demands serious consideration. The load set up by the tension of the driving chain or the

thrust from a spur or bevel gear should be taken independently of the rotor, preferably on ball bearings, in order that the friction and power losses may be confined to the sealing surface only.

If the load on the sealing device can ultimately be reduced to a value which demands a driving torque less than that which is at present required for the conventional camshaft (and with further research this objective will certainly be reached), then the gears will be very much lighter than at present found necessary for cam-operated valves. At the same time, this lightening will be possible with a satisfactory factor of safety and without exceeding the normal working stress of the material from which the gears may be made. The lower load on the teeth with negligible shock factor will conduce to appreciable savings in weight and possibly permit the use of plastics for certain components which at present are treated with such respect that only materials of highest stress value are considered by designers.

Compression Ratios

In addition to the points of outstanding merit which have been fully discussed, namely silence and non-reciprocation, the rotary valve engine possesses a third valuable feature in being able to work with an extraordinary high compression ratio, and also with low-grade fuel, down to 66 octane value, without pinking or knocking. All the evidence goes to show that compression ratios can be used of from two to three atmospheres higher than in most engines, by the substitution of the rotary valve in place of the poppet-valve.

This valuable characteristic is brought about not so much by the functioning of the valve as from the resulting shape of the head and the elimination of the nearly red-hot poppet exhaust valve, together with a much more uniform temperature in the walls of the combustion chamber. Even the surface of the rotor in contact with the flame of explosion is continually moving, and, therefore, presenting a fresh surface to the charge throughout the firing stroke. The total effect results in a warm charge without any small pockets of mixture at a temperature greater than that prevailing in the general mass.

The real technical advantage to be derived from the use of the higher compression ratio is the increase in thermal efficiency. This in turn leads to improved fuel consumption, better performance and a reduction in the size of the required cooling radiators.

It is known that in a normal engine with a compression ratio of 6 to 1, some 70 per cent of the heat value of the fuel is lost in the exhaust or dissipated in the water jacket, but with a compression

ratio of 9 to 1 the heat losses are reduced to 65 per cent, so that for an increase of three atmospheres in the compression there results an improved thermal efficiency of 5 per cent. This is the theoretical gain, but it has been shown to be much greater in practice. This does not imply that the compression pressure can be raised indefinitely, as when a ratio of about 11 to 1 is reached the gain in efficiency, as shown by the curve in Fig. 30, becomes relatively less, and other practical considerations render it necessary, under the present state of knowledge, to work at a compression ratio not greater than 12 or 13 to 1, although in at least one instance 15 to 1 has been employed.

A fuel consumption of 0·4 lb. per b.h.p. hour has been achieved with a rotary valve fitted to an unboosted engine, and this figure is being steadily improved as research work continues. The use
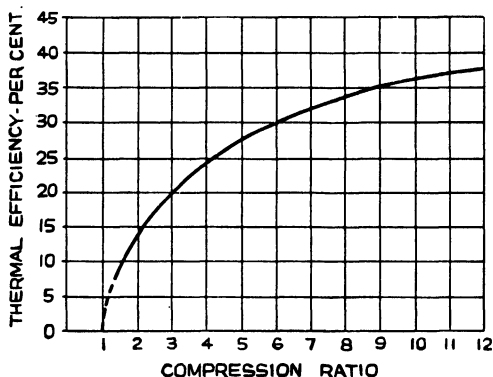
FIG. 30.
Variation in thermal efficiency with compression ratio for internal combustion engines.

of low-grade fuel and, at the same time, economical consumption are both important benefits not to be despised. The ability of an engine to run on any grade of petrol without being sensitive to variations in octane value is of commercial advantage even if there is no material difference in cost between the different grades of fuel.

Ignition

The introduction of the rotary valve leads to important differences in the accepted standards of spark timing and ignition apparatus. Firstly, the ignition point for greatest b.m.e.p. appears to be more critical than is the case with a poppet-valve engine. Secondly, it is found that a comparatively small advance is necessary to obtain the best results. Even in engines which develop their

maximum horse-power at high rates of revolution, 7000 r.p.m. or more, it is often found that the optimum point for ignition is not more than 15° before top dead centre, and 30° may be considered unusual in a high compression engine. These characteristics are brought about almost entirely by the increased compression and higher peak pressures found possible in rotary valve engines.

To appreciate fully the new conditions which follow as a corollary to increased compression it is important to note that the volume of the burned gases left in the cylinder at the end of the exhaust stroke is a smaller proportion of the swept volume than in cases where the compression is low. The products of combustion, which contain carbon dioxide and water, act as a diluent to the incoming live charge, and have a considerable influence on the
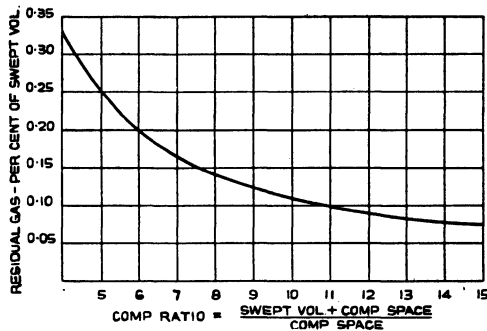
FIG. 31.
Variation of residual gas volume with different compression ratios.

rate of flame propagation and flame temperature, and upon the time taken to reach the point of maximum explosion pressure after the spark has been initiated. Therefore, the magnitude of the angular movement of the crankshaft between these two points is affected.

The extent of dilution of the fuel-air mixture can be judged by reference to Fig. 31, which shows the varying volume of the residual gases with different compression ratios. It will be seen that the residual products of combustion at 7 to 1 are approximately only half that for a compression ratio of 4 to 1, and the diluent is diminished considerably at 14 to 1, and in consequence contamination is of still less significance. It may be mentioned in passing that a specific fuel-air mixture, including the diluents, is, therefore, richer in the higher compression equivalent, and this accounts for the practical possibility of using leaner mixtures with consequent reduced specific fuel consumption.

The point of ignition, or the timing for the spark, is influenced by the rate of flame propagation and by the rate at which pressure

rise takes place. Pye* has given typical oscillograph curves (see Fig. 32) indicating the characteristics for compression ratios of 4 to 1, 5 to 1, and 6 to 1, from which it will be seen that the time and the angular travel of the crankshaft between ignition-point and maximum pressure is reduced as the compression ratio is increased, necessitating less advanced timing in order to gain the optimum pressure early on the downward stroke of the piston, i.e. soon after T.D.C. The point of peak pressure for best results is generally found to be between 5° and 10° after T.D.C. The graph given in Fig. 33 indicates the degree of ignition advance for various compression ratios. It will be noticed that the advance necessary for a compression ratio of 10 to 1 is of the order of 14°
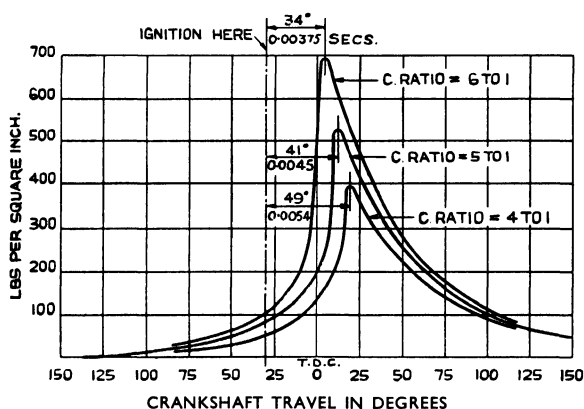
Fig. 32.
Oscillograph curves indicating characteristics for different compression ratios. (Pye.) Ignition advance 30° before T.D.C.

before dead centre. This agrees more or less with satisfactory practice.

Regarding the sensitivity of the rotary valve to variation in ignition timing, the usual range of advance and retard provided by the ordinary automatic mechanism is in most cases unsuitable. Fixed ignition, if the correct point is carefully found, gives the most satisfactory results. A fixed ignition at 10° before T.D.C. is not abnormal. Several reasons have been suggested for the critical point for ignition. The most likely reason is the extreme compactness of the combustion head, brought about by the position of the port and the high compression ratio found possible. Another reason is the shorter distance the flame front has to travel to ignite the whole charge. All timing points are brought into closer relationship, and consequently the time intervals are smaller.

Mr. R. C. Cross has recorded an interesting case of one par-

* *The Internal Combustion Engine*, vol. 1, p. 144. Pye.

ticular engine he made, which gave the best results when timed to ignite the charge at top dead centre. The reasons for this phenomenon were not discovered, but the facts are, the engine was of small capacity, only 250 c.c., and it was designed with a very high compression ratio. It has been suggested that this engine was just a freak. This will be readily admitted, but the implications are that some unknown conditions exist in the design and set up certain consequences which are the cause of the phenomenon. If the unknowns can be identified, the unusual becomes commonplace, and may be repeated if desirable in future designs.

A point of great significance is that coil-ignition has not yet been used with much success on rotary valve engines, but the
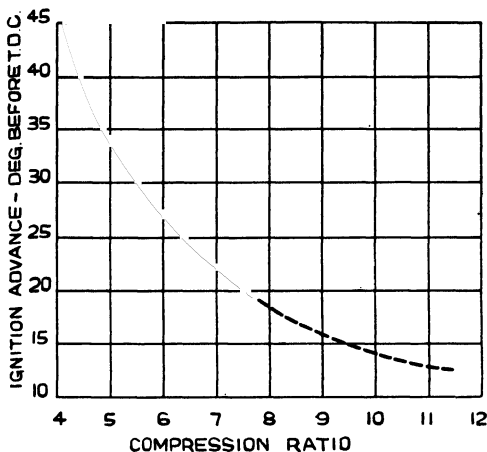
FIG. 33.
Degree of ignition advance for various compression ratios.

standard magneto gives excellent results with normal plug gaps. However, it should be recognized that there is no mysterious difference of a fundamental character between magnetos and coils, as both depend on similar electrical action. The primary difference and the cause of the different characteristics is the mode by which the current is produced in the primary winding. In the magneto the current is generated by rotation of the armature in a magnetic field, or vice versa, whilst in the coil the current is supplied from a battery of more or less constant voltage. If the essential parts are properly co-ordinated, a specified duty can be successfully met by either a magneto or an ignition coil of suitable design.

It is generally agreed that it is the core, or, technically, the capacitance component of the spark, which produces ignition. The

flame which follows and appears to surround the core is termed the inductance component, and the latter is not considered by some authorities to be of great importance. Spark failure or suppression is generally the result of either excessive leakage across the inside of the insulator or due to insufficient voltage to break down the gap between the plug points, the voltage varying with the pressure of the gases between them.

It can be said that as soon as the sparking voltage of the gap is reached, the spark is produced, and the quantity of electricity first discharged is that which has accumulated during the interval between the interruption of the primary current and the first appearance of the spark. It is this discharge that produces the capacity component of the spark. If discharge once occurs in a suitable explosive mixture, ignition should follow regardless of the inductance component of the spark. What does matter is the energy-content of the capacity component, and this can be expressed by $\frac{1}{2} CV^2$, where C is the capacity associated with the spark circuit, including plug cables and secondary winding of the spark generator, and V is the sparking voltage. As the energy-content varies as V^2, V is much more important than any variation of C. Now V depends primarily on the width of the plug-gap, and the gas pressure. It is the last named factor which creates a difference in the sparking results in a high compression rotary-valve engine in comparison with a conventional poppet-valve engine.

In the design of spark generators it is the aim to produce the highest possible value of V that is compatible with ability to overcome normal plug leakage and high compression. Plug leakage is caused chiefly by surface contamination resulting from combustion or condensation, and the effect is to lower the voltage obtainable at the plug electrodes with a given current in the primary winding of the spark generator. When testing either coils or magnetos it is necessary to simulate the plug leakage, and manufacturers have to decide on a value which is comparable with that which may exist under adverse normal conditions, due to moisture or fouling of the insulation. It is usual in such tests to employ a sparking voltage of 8000V for the test gap, which is shunted by a leak of not less than five micromhos. The test is carried out at speeds up to the maximum number of sparks per minute to meet the required duty.

It is now possible to examine the difference which exists in the characteristics of the magneto and the coil. In Fig. 34 J. D. Morgan* gives the result of a test on a magneto and an ignition coil, both designed and made to meet the normal requirements of poppet-valve engines. To obtain the curves each generator was connected

* J. D. Morgan, "Ignition Apparatus". *Proc. I.A.E.*, vol. 17.

in turn to a gap set to spark at 8000 volts, and the leakage required just to suppress sparking was found at different speeds. It will be noted that with the magneto at 200 r.p.m. a leakage of 6·2 across the test gap was required to suppress the spark. The coil required a leak of 6·5. At speeds less than 200 r.p.m. the performance of the magneto rapidly falls, while that of the coil is maintained and slightly improved. With increase of speed the performance of the magneto exceeded that of the coil, and at ordinary working speed was greatly superior as regards its ability to overcome insulation leakage.

The standard coils and magnetos in general use at the present
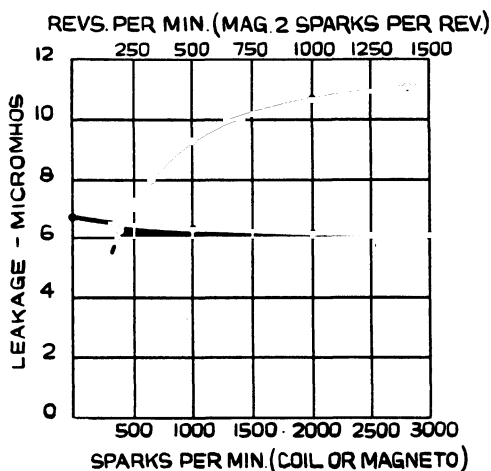
FIG. 34.
Sparking characteristics for magneto and coil. Sparking voltage 8000. Magneto advanced.
(*J. D. Morgan.*)

time have all been developed to do their work on engines with normal compression ratios up to about $7\frac{1}{2}$ to 1. For this duty, comparable operation in service has been enjoyed, but the large margin available with the magneto at high speeds enables the magneto to meet compression ratios much in excess of $7\frac{1}{2}$ to 1, whereas the falling characteristic of the standard type of ignition coil is unsuitable for the high gas pressure and number of sparks per minute required with rotary valve engines.

On technical grounds there does not appear to be any valid reason why a coil should not be designed to meet the requirements of the rotary-valve engine. When sufficient demand exists to warrant the development, there is little doubt that a coil with suitable characteristics will be forthcoming on a production basis.

CHAPTER V

EARLY EXAMPLES OF ROTARY VALVES

MANY rotary valves have been made, patented or proposed during the last fifty or sixty years, and these valves have been applied to stationary gas engines, motor-car engines, steam locomotives, compressed-air systems and to hydraulic work. In the last two applications the conditions under which they have to operate are less onerous than those present in applications to the internal combustion engine.

Only a small number of the proposed designs have actually passed through a manufacturing stage, and some only on a comparatively small scale. It is not possible to review all the inventions, but a broad selection has been made for purposes of description and illustration.

Each of the examples examined possesses some particular feature, principle or fault that invites useful discussion, and will provide a line of thought for improvement or further development in the light of modern materials, superior means of precision manufacture, and from the standpoint of greater technique in the principles of design.

The selected examples are taken in approximate historical sequence as far as records can be verified. This arrangement will afford some idea of the trend in development in relation to the availability of improved materials and manufacturing facilities at the period of individual introduction.

Rotary Valve on Crossley Gas Engines

Between the years 1886 and 1902 Messrs. Crossley Bros., Ltd., of Openshaw, Manchester, used rotary valves on both vertical and horizontal gas engines of their manufacture working on the Otto cycle. It is safe to say that this is the first occasion on which a rotary valve was applied to internal combustion engines on a substantial production basis, and a manufacturing run of sixteen years indicates that during that period the application was accepted by engineers as a sound mechanical device. These engines were provided with tube ignition and not fitted with the bare flame type of ignition which had at an earlier period been used by them in conjunction with a flat slide valve. The excellent drawing reproduced in Fig. 35

shows a section through the Crossley 4·5″ bore, engine, and gives in great detail the arrangement of the ports in the 3″ diameter
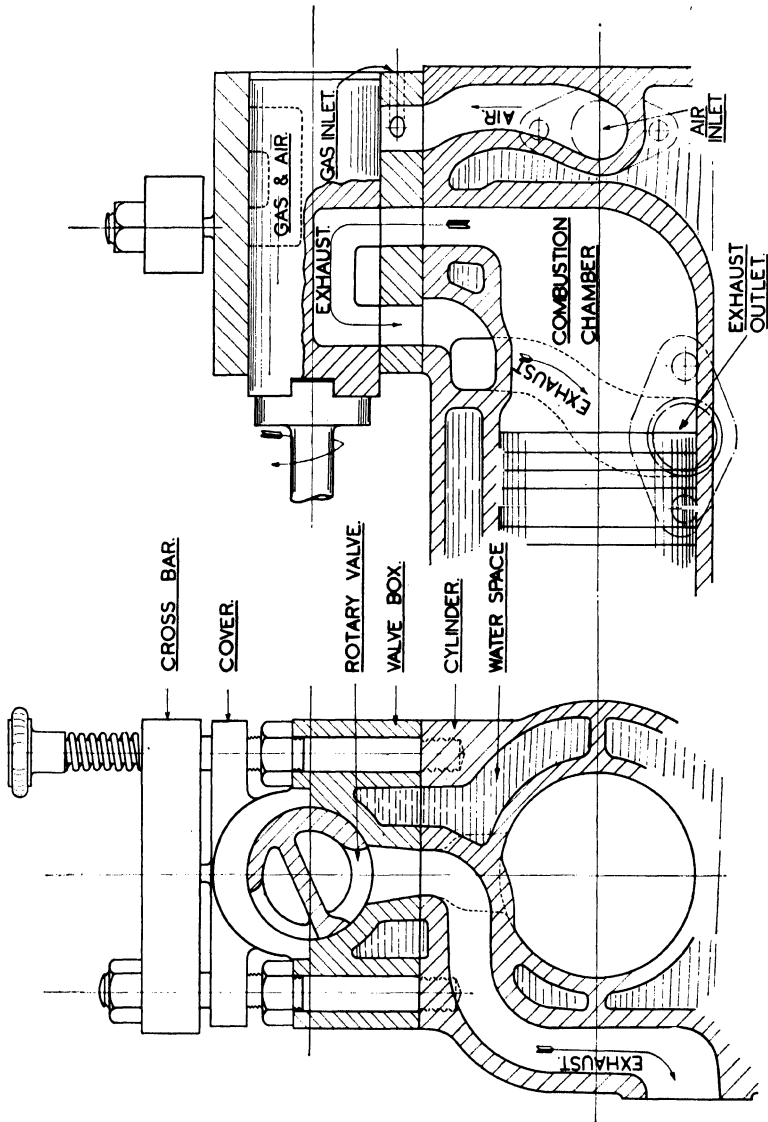
Fig. 35.
Assembly of early Crossley gas engine with rotary valve.

rotary valve through which the gas and air were admitted to the engine cylinder and through which the exhaust was expelled to atmosphere. An authentic description of the valve gear is available, and this is presented in the manufacturers' written word:

"The rotary valve was a development of one of the principles employed in the flat slide valve as used on the Crossley Atmospheric Engine and Crossley Four-cycle Gas Engine. On vertical engines the rotary motion was provided by a shaft directly driven from the crankshaft, but on the horizontal types the rotary valve itself was directly connected to the sideshaft. The mechanism, which was attached at the end of the cylinder, consisted mainly of:

(*a*) The valve box containing gas, air and exhaust passages to the rotary valve chamber in addition to a connecting passage from the valve chamber to the cylinder combustion chamber. This valve box was directly attached on to the cylinder casting.

(*b*) A valve-box cover which closely fitted over the rotary valve and into the valve box, thereby forming part of the valve chamber.

(*c*) A spring-loaded crossbar with "keep" for holding the valve-box cover in place.

"The valve establishes communication between the air inlet passage, the gas port and the cylinder inlet passage during the suction stroke, thereby allowing a gas-air mixture to be drawn into the cylinder. On the exhaust stroke the rotary valve position is such as to connect the communicating passage from the combustion chamber with that of the exhaust port in the valve box.

"Tube ignition was employed, and unlike the earlier slide-valve engine this mechanism was independent of the slide motion. The amount of gas admitted through the valve box to the rotary valve was controlled by the governor connection to a small gas valve situated near the tube igniter.

"Since the rotary valve consisted of a long cylindrical casting with several ports, perfect fitting was ensured by grinding and hand-lapping the valve into the valve box and cover. Springs were fitted to the external crossbar to assist in the maintaining of a pressure joint between the rotary valve and its cover. Furthermore, the crossbar arrangement provided an easy means of access to both valve box and cover."

It will be appreciated from the makers' description that the design of the Crossley rotary valve provided for a correct mixture of gas and air, and the arrangement in this respect is somewhat greater in scope than is at present necessary on a modern petrol engine fitted with ordinary means of carburation. The material employed for the valve and the valve box was ordinary cast iron as used generally for cylinders and all kinds of machinery at the period. Regarding the actual running of the engine, the author

is able to provide first-hand information, as fortunately about the year 1903 he had in service a small vertical engine, made by the Crossley firm, driving several machine tools in a private workshop. An old illustration of the particular engine is reproduced in Fig. 36. Although the engine was several years old, it would toil away hour after hour with an occasional application of ordinary cylinder oil to the rotor, oil probably of an inferior grade by
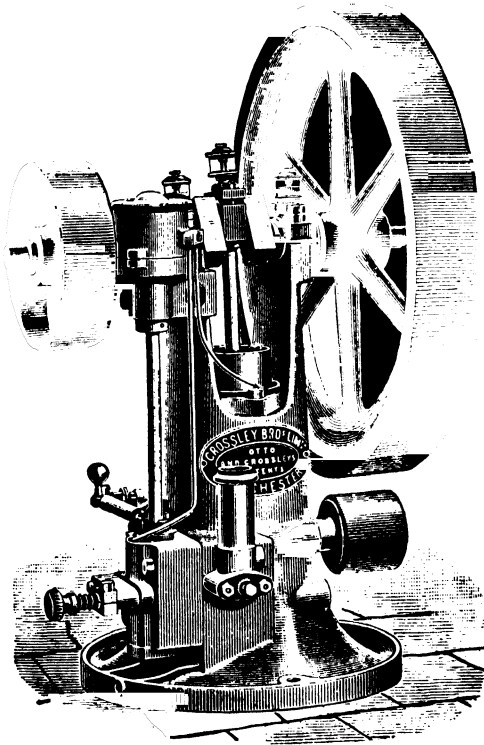
FIG. 36.
Early Crossley vertical gas engine with rotary valve.

modern standards. There is no doubt that the rotary feature was in many ways a most reliable device; that is to say, there was never a mechanical failure of any kind. On a few occasions the rotor would show signs of overheating, and in these circumstances all that was necessary to regain normal running was to remove the keep from the rotary member, clean the working surfaces with paraffin and start up again with a fresh application of clean oil. It will be seen from the illustrations Figs. 35 and 36 that the keep is spring loaded and adjustable. The operation of adjusting the spring had to be most carefully carried out. The exact loading

needed to be such that at maximum engine load, and when thoroughly warmed up, the valve was just on the point of spitting at each explosion. If the spring load was too light, the oil would be blown out each time the engine fired, and if too heavy, any extra expansion of the rotor due to increase in temperature resulted in so much additional friction that the ill-effects became cumulative and the engine would begin to lose speed. However, in spite of all this, the Crossley product was a practical engineering job, and there are many modern appliances which give far more trouble.

It is not difficult at this date to criticize the design; for instance, the arrangement for water-cooling the rotor, seen clearly in the cross section of the cylinder in Fig. 35, is only provided on one side of the box that housed the rotor, whilst the keep-side, which takes all of the thrust load from the explosion pressure, has no cooling jacket whatever. Referring to the three major problems enunciated in Chapter II, for which solutions must be forthcoming if a design of rotary valve is to endure, it is well to examine analytically the Crossley design and see how far the requirements have been satisfied.

Firstly—the question of a sealing device to ensure gas tightness. In the Crossley engine the sealing function has been secured by superfine workmanship and brute force. The load on the keep must be of sufficient magnitude to hold the rotor close up to the ported cylinder against the compression and peak explosion pressures. The bearing load on the surface of the rotor due to the spring-loaded keep is persistent throughout the whole period of each and every revolution, and there is no relaxation when the pressure in the cylinder is low or negative in value, as on the exhaust and induction strokes, whereas a self-sealing device working on the principle of the piston ring is heavily loaded only during such part of a revolution as the cycle of pressure may demand. Moreover, the superfine workmanship produced by grinding and lapping only ensures a 100 per cent seal at a specific temperature.

Secondly—in reference to friction, lubrication and power-losses, effective lubrication of a rotating member is not possible unless there is a definite circumferential clearance for the oil, although the space may be as small or even less than one thousandth part of an inch. With the close-fitting spring-loaded keep and constant dead load as incorporated in the Crossley designs, the oil wedge theory of lubrication is impracticable, and at best only semi-boundary conditions of lubrication can exist.

The general result of the spring-loaded feature is such that, in some circumstances, the friction coefficient may be as high as 0·100, whereas under more efficient hydrodynamic conditions of lubrication the friction coefficient might be as low as 0·010, or ten times less. Moreover, the area under load in this design is excessive,

and might well have been reduced by spacing the several ports closer together without sacrifice of intake and discharge areas. The need for and importance of designing for a small projected area of the rotor is greater in this system than in others where there is no spring load, for reasons which will be outlined. In the not unlikely event of some small leakage occurring round the port, the pressure of the explosion will spread over, say, one-half of the projected area of the rotor. Taking this condition as a basis, it is possible with some approximation to compute the design-load required on the keep to maintain the seal and so prevent
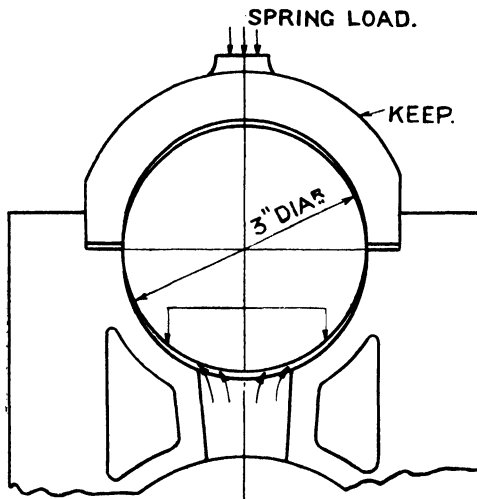
FIG. 37.
Exaggerated condition of a cylindrical rotor with expansion in a split housing.

blowing off. Assume a rotor of 3 in. diameter by 6 in. long to be subjected to a peak explosion pressure 200 lb. per sq. in., and acting on the assumed half of the projected area of the rotor bearing surface, as indicated by the arrows on Fig. 37, then the necessary load on the keep will be:

$$3 \times 6 \times 200/2 = 1800 \text{ lb.}$$

If the keep is tightened up to this load there will be an equal reaction on the opposite side between the rotor and the valve box, that is to say 3600 lb. in all. Now, if a friction coefficient of $\mu =$ 0·050 is taken and the surface distance travelled in one revolution is

$3 \cdot 14 \times 3/12 = 0 \cdot 79$ ft., then it is possible to say that the amount of work done per revolution of the rotor is:

$$0 \cdot 79 \times 3600 \times 0 \cdot 050 = 142 \text{ ft. lb.}$$

Taking the speed of the engine as 300 r.p.m., the rotor at half speed will revolve at 150 r.p.m. Therefore:

$$\text{Power loss} = 142 \times 150 = 2130 \text{ ft. lb. per minute.}$$

Or, in terms of horse-power, $2130/33{,}000 = 0 \cdot 64$ h.p. approximately. Although the data assumed may not be strictly in accordance with facts, this is a very high mechanical loss for a small engine developing perhaps two or three i.h.p. That is not all; unless this wasted heat is dissipated by the cooling arrangements, the temperature of the valve will steadily continue to rise, with cumulative ill-effects on the lubrication.

Third'y—considering the question of thermal expansion and distortion, the condition of expansion of the rotor due to a rise in temperature is, to some extent, allowed for by the introduction of the spring on the loading-bar of the keep. This, however, is not strictly correct, because if the rotor is closely fitted to the bore of the valve box when the engine is cold, which in practice would necessarily be the case, when expansion of the rotor takes place following a rise in temperature under working conditions, then the increased diameter would be nipped at the sides along the junction of the split valve box as shown exaggerated in Fig. 37. In fact, therefore, wear must take place before a stage of normal working clearance is reached.

On the basis of the thermal coefficient of expansion for ordinary cast iron given in a previous chapter, the difference in diameter of the rotor, between the cold and hot conditions, would possibly amount to 5/1000 of an inch, assuming a moderate temperature difference between the rotor and the water-cooled chest. An increase of this magnitude is quite sufficient to create the adverse set of conditions suggested.

No real provision in the design has been made to meet the possibility of local distortion due to unequal heat. In the event of either the rotor or the valve box becoming distorted due to unequal sections in the material or differences in temperature at diverse points, the original sealing fit of the valve is impaired and the super workmanship is of little account. There can be no doubt that this distortion actually takes place, as confirmed by the author's experience in the running of one of these engines under normal service conditions. A similar local distortion can and does occur at the seatings of poppet-valve engines, when full attention

has not been paid to symmetrical cooling or to correct streamlining for the flow of the exhaust gases. This distortion is sometimes sufficient to fracture the casting.

National Gas and Oil Engine Co., Ltd.

It is of great interest to know that the National Gas and Oil Engine Co., Ltd., of Ashton-under-Lyne, after a search through their early records, are able to confirm that two series of gas engines with rotary valves were produced in the early days of the company. The first series was manufactured in 1895. The only working drawing of this engine in existence at the present time has been reproduced from the original and is shown in Fig. 38.

It will be noticed that the intake valve only is of rotary form, the exhaust valve being of the poppet-type actuated by a rocker from a cam mounted on the same side-shaft which caused the rotor to revolve.

Unlike the Crossley arrangement, the valve chest is not water cooled, and it is placed some distance from the cylinder barrel with a comparatively long communication duct between the rotor and the combustion head. There is sufficient detail on the drawing, although not of great clarity, to indicate that the rotary valve works in a one-piece bored chamber which is not split, and consequently an exact working clearance had to be produced by fine workmanship in the first place and without means of later adjustment. The dimensions of this engine are given: bore of cylinder 5·5 in.; stroke 10 in.; approximate horse-power 4 at 260 r.p.m.

The second series of engines differed materially from the first design in having a single rotary valve controlling both the gas-air intake and the exhaust without any poppet-valves. Details of the rotor and the valve box with the leading dimensions are indicated in Fig. 39. Unfortunately, no assembly drawings of this engine are now available, but it is known that the dimensions of the cylinder are somewhat smaller than those of the first series, the bore of the smaller engine being 4·5 in. and the stroke 8 in., approximate horse-power 1·5 at 300 r.p.m. It can be assumed that the compression ratio on engines of both series would be moderate and the valves some distance from the combustion head, so that the temperature rise would be somewhat gradual after starting up, and therefore both rotor and valve box would have the opportunity to expand at an even rate in response to thermal changes, and it is hardly to be expected that the valve and chest would reach a very high temperature. As previously stated, the valve box was not split, so it is clear that the rotor worked with a defined nominal annular clearance. The valve was therefore working
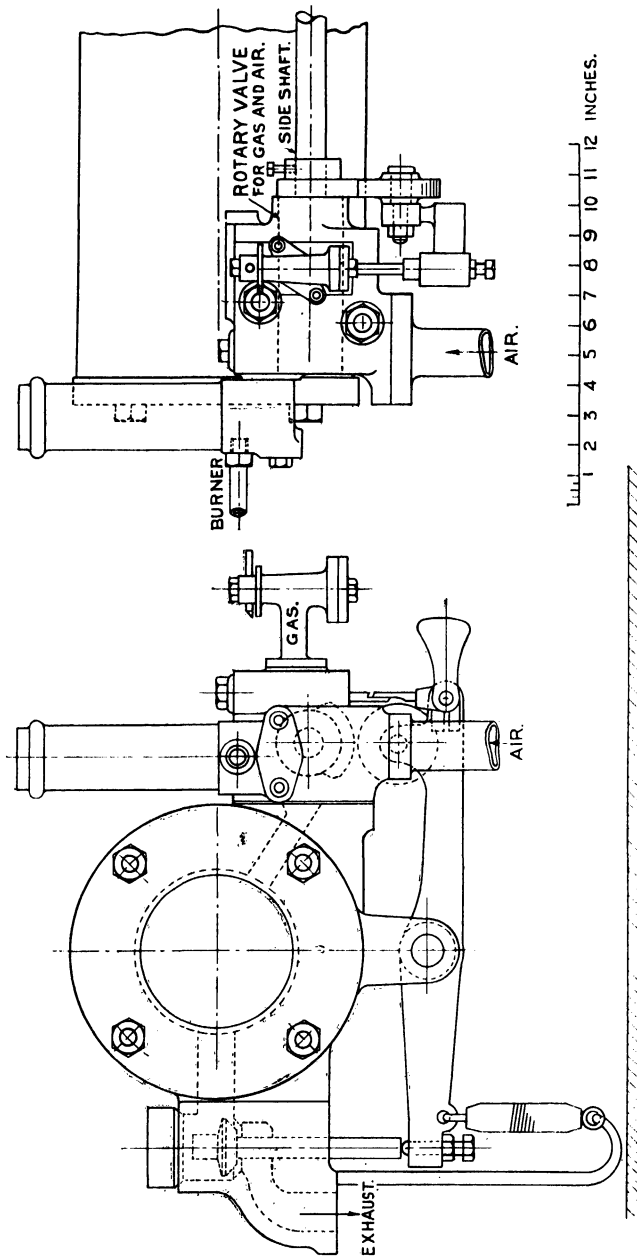
Fig. 38.
Assembly of early National gas engine with rotary inlet valve. First series

nearer to ideal hydrodynamic conditions of lubrication than are likely to exist in an arrangement embodying a spring-loaded keep as employed by Crossley.

Possibly the seal was far from perfect until the valve had properly warmed up, but the friction losses would most certainly be moderate under all conditions of temperature. The approximate order of the power-loss can be readily computed by evaluating the thrust caused by the compression and explosion pressures acting over the area of the port aperture, plus a proportion of the projected area of the rotor.

The full scheme of lubrication is not disclosed, but an oil channel
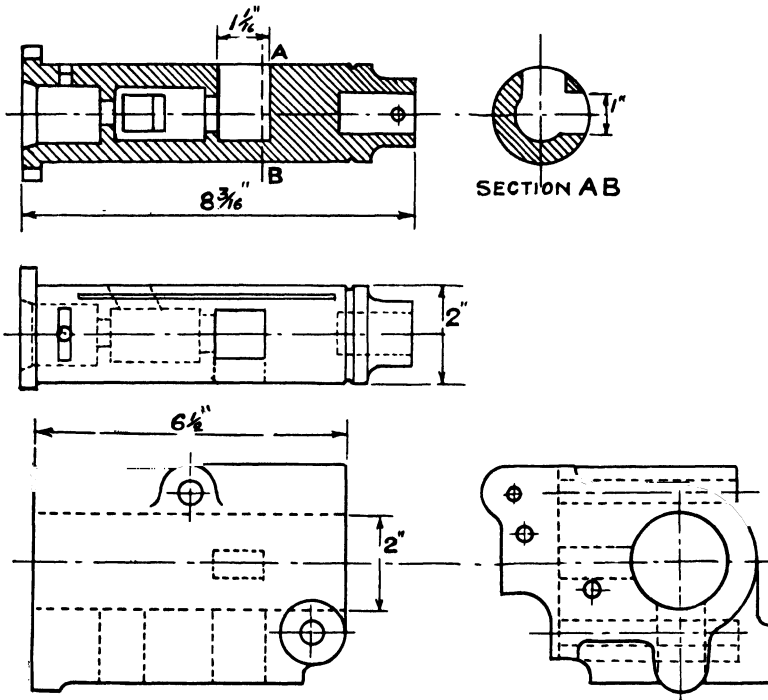
FIG. 39.
Combined inlet and exhaust rotary valve details of an early National gas engine. Second series.

is clearly specified on the detail drawing of the rotor, and this presumably was fed with oil from a drilled oilway in the valve chest. Both of the engines described above were manufactured in considerable quantities and could not have failed to provide the users with genuine satisfaction. This view is more or less confirmed, since the engineers of the National Gas and Oil Engine Co., Ltd., have stated that some of these rotary-valve engines were being serviced up to the year 1935, which leads to the conclusion that the rotary valves were produced and serviced by this company during a period of over forty years.

Reference has been made in Chapter I, under the heading "Historical Foreword", to a statement made by the late Sir Dugald Clerk at a meeting of the Institution of Automobile Engineers in 1911,* when he said that the National Gas Engine Company had several years previously experimented with a centre plug valve surrounded by a jacket or casing through which the exhaust gases, instead of water, were circulated in order to ensure that the rotor and the valve seating both expanded together at the same rate. This statement is both interesting and important, as such an arrangement in some measure exemplifies the principle of a rotating valve working with a constant annular clearance during all thermal variations and conditions from starting up to full load, provided that the temperatures of both components are controlled by some means to rise and fall in unison. It should be noted that the principle was not only envisaged, but that convincing proof is recorded that the valve worked well on an experimental engine. It was stated to have run exceptionally well at about 1000 r.p.m. and it was in service in the works for a period of two years without any trouble. The Company have at the suggestion of the author recently carried out an extensive search for data and drawings of this engine, but it is to be regretted that no records have been unearthed that provide any further information on this valve gear.

It is of importance and of value to bear this principle in mind, as the use of present-day special materials may enable success to be achieved by taking advantage of differential expansion.

Mr. C. Lorenzen's Rotary Valve With Balanced Pressure

Mr. C. Lorenzen made a small single-cylinder engine about the year 1906, employing a rotary valve with a number of original features, and it was stated to have worked so satisfactorily that a four-cylinder engine of 30/40 h.p. was built on similar lines. The larger engine was brought to this country about 1909 with a view to placing it on the market, but little more was heard of it. The arrangement of the valve gear is shown in Fig. 40. Two definite claims were made for the design. Firstly—the difficulties of excessive friction had been overcome; and Secondly—leakage was prevented by the provision of a seal with an automatic tightening device. The rotor consisted of a hollow cast-iron body rotating on ball bearings, water cooled from the inside and provided with suitable ports for the passage of the gases. To ensure gas tightness, a casing in two halves, somewhat resembling bearing caps, was slightly pressed by flat springs against the valve, but in order to

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* *Proc. Inst. Mech. Eng.*, 1911-12.
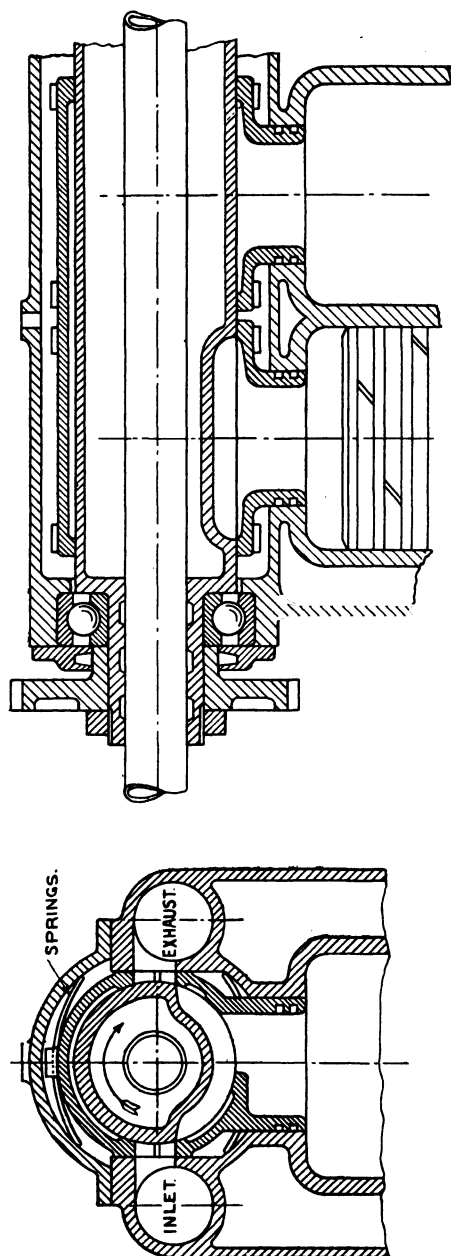
Fig. 40.
Lorenzen's rotary valve engine with pressure balanced sealing device.

minimize the friction, and to concentrate the pressure where mostly wanted—that is, round the cylinder ports—the latter have projections or lips round them which alone press against the rotary valve

in the same way as piston rings do against cylinder walls. The cavities or reliefs between these projections form lubrication reservoirs. The lower shell, which is made in four separate parts in the case of a four-cylinder engine, is also provided with extensions in the shape of pistons, fitting into a central bore in the head of each cylinder, and made tight by means of ordinary compression rings. The central part of the piston forms a port to allow the passage of the gases, widened, however, on the valve side to nearly a rectangular shape, with an area of about the same size as is the area of the piston part on the under-side. In consequence of this arrangement, part of the gas mixture will be compressed and exploded in this widening, thereby counterbalancing the pressure of the exploding gases on the under-side of the shell, which will therefore be pressed against the rotor by the constant pressure of the springs or by a slightly greater pressure, which in practice Mr. Lorenzen found to be desirable during the explosion stroke. By varying the ratio of the two areas to one another any desired margin of sealing pressure could be obtained.

In reviewing this design it can be said that the features described above are sound in principle. The sealing component has a strictly limited area, considerably less than the area of the main piston, and then only part of this area constitutes a load on the rotor, due to the balancing area on the inside. This is a particularly novel manner of gaining an automatic seal which can at the same time be regulated by design dimensions to provide just the necessary pressure to obtain the best results. No dimensions or sizes are, at this period, available in the description of the valve, so it is not possible to assess the power-losses quantitatively, but they would most certainly be on a moderate scale. The cavities provided to act as oil reservoirs are not considered an ideal arrangement, as on the suction stroke the excess oil would most certainly be taken into the cylinder along with the charge of gas. Had the relieving spaces been made only a few thousandths of an inch in depth and the leading edges been scraped to allow the oil wedge to function, there would have been no serious objection to their use, always provided the supply of oil was fed to the valve at a suitable rate.

There is one constructional feature in the design which demands precision workmanship of a kind not usually met with in the manufacture of rotary-valve engines, as generally almost all of the machine work is cylindrical. An inspection of the sectional view taken through the head of the Lorenzen engine will reveal that to ensure an approximate seal between the flat sides of the split bearing caps and the housing into which they fit adjacent to the intake and exhaust ducts, the need arises for good fitting flat surfaces. This means that the surfaces have to be produced by milling or planning, and the distances of all flat faces on both the

split bearing caps and the housing must be more or less equidistant in a lateral direction from the axis of the cylinder bore, in order to avoid interference with the close controlling fit of the piston portion of the seals. This is a class of machine accuracy not lightly undertaken and generally strongly objected to by the factory producing the components. If an attempt be made to avoid this difficulty by allowing wider tolerance in manufacture, then there would certainly arise the possibility of pollution of the exhaust gases by the incoming charge, which often makes starting difficult. This type of rotor is open to the same fundamental technical objection as any rotor with a single scooped-out portion functioning as a passage for both inlet and exhaust gases—that it permits the trapping and consequent wastage of live gas, and the contamination of fresh mixture by the exhaust gas.

Steam Locomotive With Rotary Valves

In recent issues of *The Railway Gazette* and *The Engineer** Mr. J. Clayton refers to the construction of an unusual design of locomotive at the Derby works of the old Midland Railway Company during the year 1908. The engine is described as of the 2-6-2 wheel arrangement and was designed and patented by Mr. (later Sir) Cecil W. Paget, then works manager. The engine had eight cylinders, single acting, 18″ diameter by 12″ stroke, operated by a unique design of rotary steam admission valves. This appears to be the first case where a rotary valve has been applied to a steam locomotive. The single-acting cylinder with its "no reversal of thrust" particularly appealed to the designer, as did the short stroke, simplifying the balancing, while multiple cylinders reduced the condensation losses. A very moderate wheel diameter of 5 ft. 4 in. was adopted in the belief that this would be no bar to high speed, and, indeed, on one of the test runs the engine attained a speed of 82 m.p.h. with its train.

In order to obtain quick admission of steam to the cylinders, Paget applied himself to a design of rotary valve, one to each set of four cylinders. This valve only controlled the admission, the exhaust occurring through ports in the cylinder barrel at the end of the piston travel, similar to the Willans stationary steam engines, and on the same principle as is used for the exhaust on many two-stroke I.C. engines of today.

In the original design of the engine the rotary distribution valves were placed vertically between the cylinders and contained in the same casting. Many tests were carried out on a full-size

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* "Links in the History of the Locomotive", *The Engineer*, March 26, 1943. "The Paget Locomotive", *The Railway Gazette*, November 2, 1945.

[By courtesy of "The Railway Gazette".

FIG. 41.

Paget's steam locomotive cylinder and rotary valve casting.

model of the valve to ascertain the tightness and the amount of wear. The engine as finally designed and built had the valve

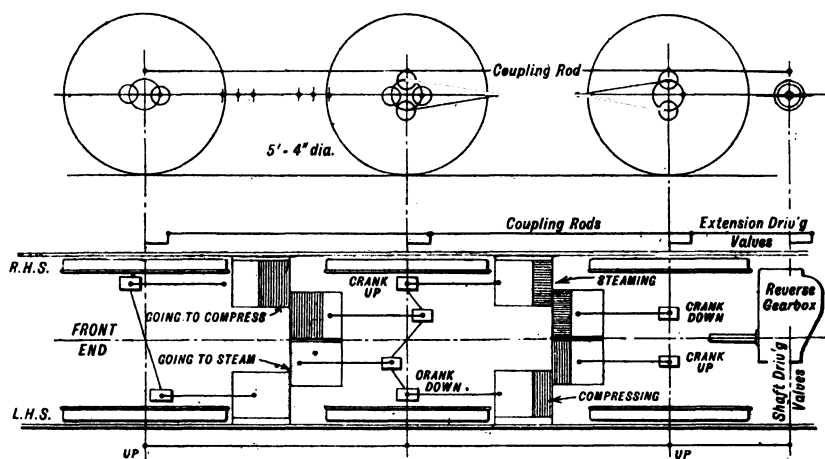
chest placed on the centre line of the engine in plan, horizontally above the cylinders and integral with the same casting, as shown in Fig. 41. This position simplified the drive by using a through shaft from an epicyclic train of gearing under the footplate, which formed the "reverse" mechanism, all being self-contained in an oil-tight casing. This train of gears was arranged in a similar manner to the differential gear used in motor-cars, being driven

[By courtesy of "The Engineer".

FIG. 42.

Diagrammatic plan and elevation of Paget's locomotive showing layout of cranks, cylinders, pistons and rotary valves.

CYCLE OF CYLINDER CONDITIONS

Position.	*First quarter.*	*Second quarter.*	*Third quarter.*	*Fourth quarter.*
L.H.F.	Going to steam.	Steaming.	Going to compress.	Compressing.
R.H.F.	Going to compress.	Compressing.	Going to steam.	Steaming.
R.H.B.	Steaming.	Going to compress.	Compressing.	Going to steam.
L.H.B.	Compressing.	Going to steam.	Steaming.	Going to compress.

by the cross shaft with cranks on each outer end, which in turn derived motion from extensions of the trailing coupling rods.

The cylinder steam chest was circular, 10" bore, and in it there turned a steam-tight fitted sleeve liner of phosphor-bronze, ¾" thick, in which ports were formed in order to provide a means of varying the cut-off point of the steam fed to the cylinder by the valve. This liner was smoothly bored, 8½" diameter by 16¼" long, to receive the valve in the form of a cast-iron split bush fitting therein. The valve, being split, was forced open by the steam pressure on

its inside so as to fit the liner. As mentioned, the valve was driven by the revolving shaft through its centre, on which was mounted a keyed sleeve having dogs engaging suitable projections cast on the inside of the valve.

The cut-off liner was controlled and adjusted by a special shaft and pinions actuated through racks by a steam and hydraulic cylinder, as used for the well-known locomotive steam reversing gear. Reverse was effected, as already stated, by gearing which was controlled by a vertical hand-operated screw, arranged in the cab beside the driver. This action turned the valve through 120°, thus altering its angular relation to the lay shaft and the ports through the cut-off liner. The diagrammatic plan, Fig. 42, shows the layout of the cranks, cylinders, pistons and valves with the sequence of steam events for one revolution.

The rotary distribution valve described proved difficult to keep steamtight, with the result that steam leaking past it found its way into the opposite port and so to the pistons on the compression stroke, causing as much backward pressure as forward. (In the ordinary engine, where the D slide or piston valve also controls the exhaust port, any leakage past the admission edge escapes to exhaust, causing little or no back pressure.) The form of rotary valve tried first on this engine was a plain cast-iron cylinder with its port way divided into three by bridges, and was split in its length in the centre of the port, so that the steam pressure inside could expand it, causing it to fit steamtight against the liner (*see* Fig. 43). As a result of further experiments the surface of the liner was relieved by flutes so that a proportion of the pressure causing the rotor to expand was balanced by steam on the outside, to the extent of approximately 40 per cent, thereby reducing the friction between the rotor and the sleeve in the same proportion. The bronze cut-off liner had two sets of ports through it, four on each side, each set feeding respectively the left- and right-hand port of the cylinders. By turning the liner it was possible to vary the number of ports open to steam according to the cut-off required; thus with all liner ports open to steam, cut-off was 75 per cent stroke; three liner ports open to steam, cut-off was 59 per cent stroke, and so on.

Other forms of rotary valve were proposed to get over the difficulty with the first type tried, but unfortunately for the engine and its further trials, Mr. Paget was appointed General Superintendent of the Midland Railway, and he became so engrossed in his new duties that nothing further was done with the engine. The actual running of the engine as built showed that it ran well and steadily at all speeds up to and over 80 miles per hour. It was also satisfactory as regards management by driver and fireman.

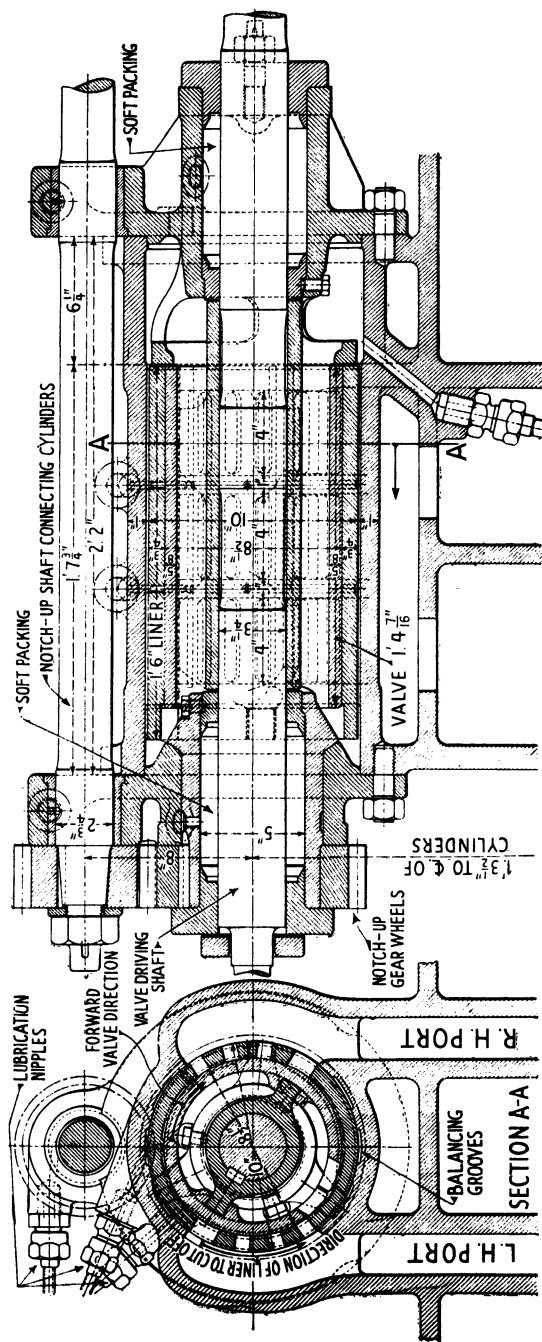
The chief cause of failure is specifically recorded by Mr. J.

[*By courtesy of "The Railway Gazette".*

Fig. 43.

Cross-sections through rotary valve of Paget's steam locomotive.

Valve has no lead.

With all liner ports open, cut-off = 75 per cent. stroke.
" 3 " " " " = 59 " " "
" 2 " " " " = 42 " " "
" 1 " " " " = 25·5 " " "

Clayton, who was closely associated with the design and construction. The types of rotary valve tried all had the same defect, namely after running at high speed excessive leakage of steam occurred at the valve port edge, round the body of the valve, and into the cylinders opposite to those which were steaming. The heat generated by friction at high speeds caused distortion; the edges of the valve, at the port, curled inwards owing to the outside of the valve becoming hotter than the inside. Mr. Clayton has suggested that in the main the chief trouble was "metallurgical", and that in these days of more advanced knowledge on the subject of high duty materials there would have been no difficulty in producing a valve and liner suitable to work on the rotary principle embodied in the Paget design.

There are some grounds for the above suggestions, but the power-losses due to friction appear to have been considerable, and were without doubt the cause rather than the effect, leading to the partial failure.

The author has had the privilege of examining the actual experimental records of the preliminary tests which were undertaken on the full-size rotary-valve unit under live steam. Two of these test sheets are shown in Tables 6 and 7, which provide full

TABLE 6

STARTING TORQUE OF ROTARY VALVE

LINER FLUTED TO BALANCE APPROX. 40 PER CENT OF STEAM PRESSURE

Test Taken February 17, 1907

Ports Open	*Steam Pressure*	*Weight in lbs. on 24" lever*	*Weight in lbs. per lb. of Steam Pressure*
1	177	168	..
2	177	391	..
3	180	480	..
3	180	424	..
Full open	177	856	$\left.\right\}$ $\frac{5380}{1245} = 4{\cdot}321$
" "	174	648	
" "	177	801	
" "	177	826	
" "	180	912	
" "	180	536	
" "	180	801	

information on the starting torque (static condition of friction) and the horse-power found necessary to drive the valve at speed (kinetic condition of friction) under several conditions of pressure and cut-off. It will be observed that the force to drive the valve is

greatest when all ports are open, as would naturally be expected, but this is a condition which obtains for short periods only, as when starting the engine from rest.

From the values given in Table 6 it is possible to deduce that the torque required to start the valve from rest under full steam pressure with all ports open is of the order of 19,200 lb. in., and by further evaluation to verify the coefficient of static friction. Making due allowance for the 40 per cent balancing action of steam on the relieved liner, as mentioned previously, a calculation shows that μ under these conditions is approximately 0·10. This figure considered only as a coefficient of static friction is not unduly high and compares favourably with established data for the ordinary locomotive D slide valve of brass working on a cast-iron face (dry) for which μ is usually taken as 0·18.

TABLE 7

POWER CONSUMED BY MOTOR DRIVING VALVE

VALVE SPLIT AND BALANCED 40%

Test Taken April 4, 1907

No. of Test	*Speed Revs. per Minute*	*Steam Pressure*	*Ports Open*	*Amperes*	*Volts*	*E.H.P.*
1 ..	214	130	1	15	314	6·32
2 ..	204	155	1	12·5	326	5·47
3 ..	224	155	1	14	322	6·03
4 ..	208	145	2	27·5	311	11·45
5 ..	210	155	2	32·5	313	13·63
6 ..	180	140	3	37·5	290	14·57
7 ..	200	95	Full open	22·5	326	9·84
8 ..	192	100	Full open	32·5	313	13·63

Proceeding next to a consideration of the power-losses as indicated in Table 7, as a basis for calculation it can be assumed that at times when the locomotive was running at a maximum speed of 80 m.p.h. the valve cut-off would be set to one port open, additional ports only being required at starting or for climbing gradients. If the recorded figures from Test 1, Table 7, are taken, namely revs. per minute 214, steam at 130, one port open, horsepower 6·32, then the last figure can be corrected for full boiler steam pressure at 180 lb. per sq. in., and maximum train speed of 80 m.p.h. At this train speed the rotary valve will be running at 420 revs. per minute. Then:

$$6{\cdot}32 \times 420/214 \times 180/130 = 17{\cdot}4 \text{ h.p. for one valve.}$$

Again, assuming a cut-off condition with two ports open for gradient climbing and a train speed of, say, 60 m.p.h., when the rotor will be turning at 315 r.p.m. and using the recorded figures from Test No. 5, namely r.p.m. 210, steam at 155, two ports open, horse-power 13·63, the horse-power can be corrected for full steam pressure and a train speed of 60 m.p.h. as follows:

$$13{\cdot}63 \times 315/210 \times 180/155 = 23{\cdot}7 \text{ h.p. per valve.}$$

This probably represents the case of the heaviest duty to which the valve is subjected.

The power-loss due to friction may alternatively be calculated from first principles on the lines indicated in Chapter II, and for this purpose the following data can be taken:

DATA: Surface area of rotor = 430 sq. in.
Nett area allowing 40 per cent balanced = 250 sq. in.
Circumference of rubbing surface = 2·22 ft.
Pressure on surface of valve = 180 lb. per sq. in.
Speed of rotor = 315 r.p.m.
Coefficient of friction—assumed μ = 0·025.

Then the work done in turning the valve against friction can be evaluated in ft. lb. per minute as follows:

$$315 \times 2{\cdot}22 \times 180 \times 250 \times 0{\cdot}025 = 800{,}000 \text{ ft. lb. per minute,}$$

hence power-loss = 800,000/33,000 = 24·2 h.p. Confirmation is here provided that the assumed coefficient of friction μ = 0·025 is a fair value to take for a rotor working under the conditions stated, as the horse-power consumed is of the same order as that assessed by using the figures given in the actual test records.

The rotary valve as fitted to the engine was lubricated from a mechanical lubricator feeding oil by way of two circumferential grooves with radial holes spaced round the bush in which the rotor revolved. The oiling system was therefore well carried out, but it is known that efficient lubrication of a continuously loaded member without nominal clearance is not an easy matter. If, as is actually the case, the temperature of saturated steam at 180 lb. per sq. in. is taken to be 374°F. (190°C.), then the generation of the heat from approximately 25 h.p. superimposed at the surface of the rotor already at 180°C. would not be conducive to efficient lubrication. It is, therefore, possible that if the valves had not curled inwards at the edges, as reported, they might later have seized with more disastrous results.

The indicated horse-power of this engine is stated to have

been of the order of one thousand and, as has been shown, the friction losses for the two valves were approximately 50 h.p., so the power losses amounted to 5 per cent. This must be considered an unduly high proportion of the i.h.p. of the engine.

Considerable space has been given to the analysis of these results in order to show that the application of a rotary valve to a steam unit is by no means an easy problem, from the fact that the sealing arrangement is never, and fundamentally cannot be, relieved of pressure between the power-strokes as is possible with an internal combustion engine.

The real defect in the design is the excessive area of the sealing device, continuously loaded under running conditions, with the full boiler pressure of 180 lb. per sq. in. and no relief with varying cut-off during periods of expansive working. It is unfortunate that circumstances prevented further work being undertaken on the sealing device, especially as the other unorthodox features in the design proved to be so successful, for which nothing but praise can be given to the designers.

Adams Compressed-Air Engine Starter With Rotary Valve

Motor-cars produced by Adams Manufacturing Co., Ltd., Bedford, about the year 1908, were fitted with a compressed-air system for starting the engine and the arrangement included a flat rotary distribution air valve which in service proved to be a very successful device.

Fig. 44 shows the distributor valve in detail. Fig. 45 indicates diagrammatically the complete starting system with the cylinder block, pipework, air reservoir and the rotary valve mounted on the crankcase. Combined with the distributor box is the starting valve operated by the driver by means of a pedal on the dashboard. The reservoir of compressed air is in communication with the pipe connected to the under-side of the starting valve. Branch pipes from the distribution chamber communicate with the separate cylinders of the engine. For starting, the pedal is depressed against the action of a spring to operate a crank which opens the starting valve. This valve consists of a ball pressed by a spring on to a spherical seating. A pin engaged by the crank depresses the ball, allowing air to pass from the feed pipe into the distributor box. This air travels to the top side of the rotary disk, which when in action derives motion from the crankshaft through the two-to-one camshaft drive.

The rotary distributing valve consists of a flat circular disk in which are one or more ports rotating over a number of ports, corresponding to the number of cylinders of the engine. The

openings in the disk and the ports are so arranged as to admit air from the chamber to the ports and thence to the branch pipes leading to the cylinders, during the explosion stroke in each cylinder in the correct order of firing. The rotor is secured to and driven by the spindle which has a bearing in the lower portion of the valve

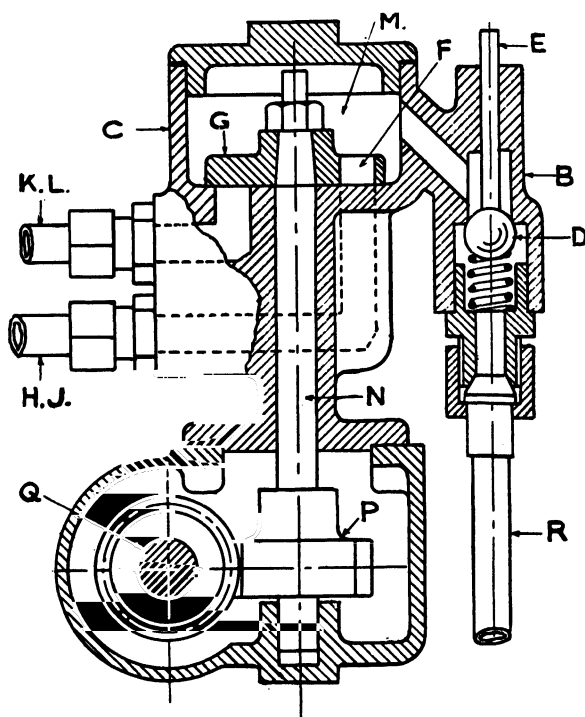



FIG. 44.
Adams pneumatic rotary distributor valve for self-starter.
B—Starting valve. C—Distributor box. D—Ball valve. E—Pin. F—Port. G—Rotary disk. H and J—Pipes to cylinders 1 and 2. K and L—Pipes to cylinders 3 and 4. M—Valve chamber. N—Valve spindle. P—Worm wheel. Q—Camshaft. R—Feed pipe.

box. As the rotor is driven from the under-side there is no call for a gland, which would be necessary if it were driven from above, that is to say from the pressure side of the valve.

The number of holes in any instance would in practice depend upon the speed adopted for the rotor and on the actual firing conditions of the engine. In Fig. 44 the spindle carries a worm wheel and this is engaged by a worm on the engine camshaft. The two gears are enclosed and protected in an oil box.

The cylinders of the engine are indicated in Fig. 45, which also shows the air pipes from the distribution box connected to non-return inlet valves, one for each of the cylinders. These check valves are intended to retain compression during the normal working of the engine.

An important feature of the system is that the rotor rests only lightly on the seating by gravity and very little friction results during the ordinary running of the engine. It is only during the short period of time when the actual starting operation is in progress that the air pressure is exerted upon the rotating member; that is, when the seal is a definite requirement.

This type of rotating valve with a system of separate connexion pipes can be employed for many purposes, including the distribution

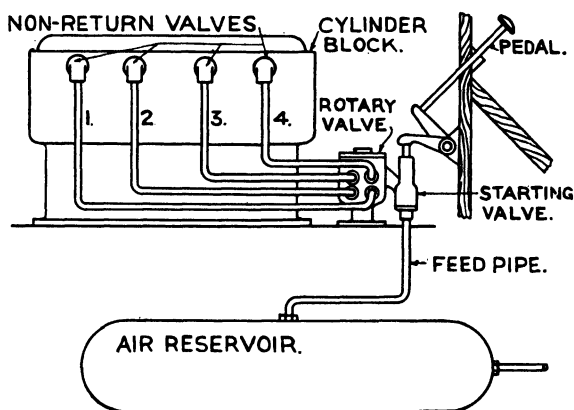



FIG. 45.
Adams pneumatic self-starter system with rotary valve applied to a motor-car engine.

of steam to the separate cylinders of a multi-cylinder engine, also for hydraulic machines using oil or water for a great diversity of functions. The system, however, is quite unsuitable for application to an internal combustion engine due to the varying length of the connecting pipes. This handicap is not of great moment in a steam engine or in hydraulics.

Speedwell Rotary Valve Engine

The illustration of the Speedwell engine with rotary valves (Fig. 46) is taken from *Audel's Automobile Guide*, published in New York in 1915–18. The drawing is in diagrammatic form, and not in great detail, so that it is only possible to discuss the design on

general premises. The description accompanying the illustration is also somewhat meagre, but it will be seen that the design is distinctive in having two separate cylindrical rotors, one for controlling the intake and the other the exhaust. The rotors, which are placed one at each side of the combustion head, are slotted right through, and in rotating the slots register with ports in the cylinder walls, thus performing their respective functions of intake and exhaust. The valve movement is a continuous revolution in

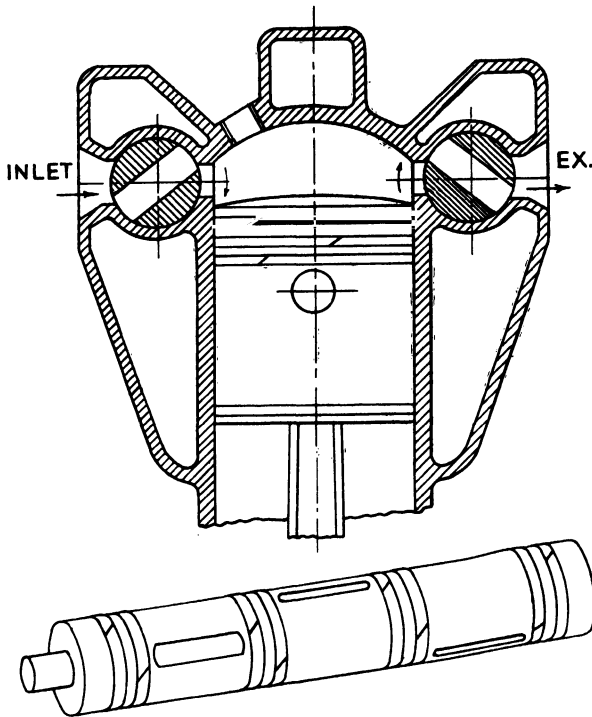



FIG. 46.
Speedwell three-cylinder I.C. engine with independent cylindrical rotors for inlet and exhaust.

one direction. The arrows within the cylinder indicate the direction of rotation of the rotor; the arrows outside indicate the direction of the live gas passing in and the exhaust gas passing out.

The above is the extent of the descriptive matter available, but it can be gathered that there is no provision for sealing between the combustion head and the rotor, and in this respect all the objections given in a previous chapter apply. It will be seen from the pictorial view of the rotor element that some attempt is made, by the introduction of compression rings, to prevent leakage between

one cylinder and another; but the rings do not obstruct the main path of leakage by way of the working clearance between the rotor and the horizontal bore which encircles it. There is no provision for thermal expansion and distortion other than by the essential working clearance, and the design is condemned on this account alone. There is nothing to indicate or to suggest that the rotor is supported at each end on independent bearings, and, lacking this information, it can be assumed that these were not included in the design. The forces due to the pressure of the gas tending to spread over the area of the rotor would result in considerable friction, although the extent might be somewhat limited by the compression rings. The thrust from the driving gear will also add its quota of friction and the power-losses under these handicaps would be high, especially in view of the fact that the losses are doubled due to there being two rotors.

There is, however, a decided advantage in separating the exhaust from the inlet. The incoming charge under these conditions is not in any circumstances subject to contamination from the exhaust products. This is a most important consideration during such times as the throttle may be nearly closed for slow running, when a possibility exists with a single rotor of direct leakage between the exhaust and the inlet due to the vacuum created in the clearance space.

The value of independent rotors with full separation of the live gas and the exhaust, as a method of preventing contamination, is not to be overlooked.

The pictorial view of the rotor component indicates that the Speedwell engine was intended to have three cylinders only, and possibly it was thought that the difficulty of machining a long horizontal bore for a four-cylinder engine would be too great a tax on the machine shop, in view of the high degree of accuracy necessary for a design of this kind. However, the small difference in length between a three- and a four-cylinder engine, in itself, would not combat the difficulties of thermal expansion and distortion, which are not to be overcome by precision manufacture. It is quite possible this design never advanced beyond the experimental stage.

A system comprising twin rotary valves, however, must not be prejudged on the score of unnecessary duplication of components, as it is noteworthy that any arrangement with separate rotors for intake and exhaust has a most important influence on the problem of lubrication. A valve chest of the exhaust-cum-inlet type of rotor is alternately subject to a pressure and a depression. This condition results in a pumping action on the lubrication supply, which tends to draw in an excess of oil during the suction stroke when it is least required, and tends to force oil away at other parts

of the cycle when extra oil would be of most benefit. In order to overcome this contrary state of affairs it is invariably necessary to introduce into the oil system some ancillary apparatus which will overcome the handicap.

This situation does not occur when the inlet and exhaust rotors are separate, as there is then always a depression in the intake-valve chest and a modulated pressure in the valve chest of the exhaust. It will therefore be evident that each of the rotors can be treated separately and the quantity of oil regulated according to the needs of one and the other.

Russel Dual Rotary Valve

The Russel Dual Rotary Valve is in some respects similar to that used by the Itala Co. of Turin, and referred to under that heading on a later page. The Russel engine was built in the United States of America for a number of years commencing about the year 1911 and was used principally on farm power plants and lighting sets. Fig. 47 is a sectional view of the cylinder, valve chamber and the rotor of the Russel engine. A single valve rotating at one-fourth crankshaft speed serves two adjacent cylinders. The valve is tapered and it turns in a bore of corresponding form. Passages for both inlet and exhaust extend diametrically through the valve, opposite ends of these passages registering with the cylinder port at intervals of one cycle or two crankshaft revolutions. Gas enters the valve chamber at the top, while the exhaust escapes through another port in the side of the valve chamber near its bottom.

The valve is driven by a special mechanism. Its lower cylindrical part carries a pair of horizontal pins with rollers extending radially inward. The rollers are located in double helical grooves in a collar at the upper end of the valve drive shaft. The collar has two radial lugs on it, and pins extending parallel with the axis of the collar are fastened centrally into the lugs. These lugs are engaged by two coil springs under compression, which also bear against two keys, integral with the lower part of the valve. Owing to the interaction between the helical grooves and the roller pins, pressure exerted by the springs tends to draw the valves against their seats. With the engine at rest the valve is seated tightly. When the engine is cranked or is turning under power the valve shaft travels ahead of the valve by a fraction of a degree and, on account of the action of the helical grooves and roller pins, raises the valve slightly in its seat, thereby freeing it. Thereafter the valve automatically maintains the proper seating condition, rising when it becomes tight and vice versa.

In this design the two ports in the cylinder wall (corresponding to adjacent cylinders) are spaced at 45° from centre to centre. Since the valve rotates at only one-fourth crankshaft speed, to pass from a given position with respect to one cylinder port to the same position with respect to the other cylinder port takes the time of 180° of crankshaft rotation or one piston stroke, and the cranks of the two adjacent cylinders are set 180° apart.

The arrangement of the Russel engine is of particular interest

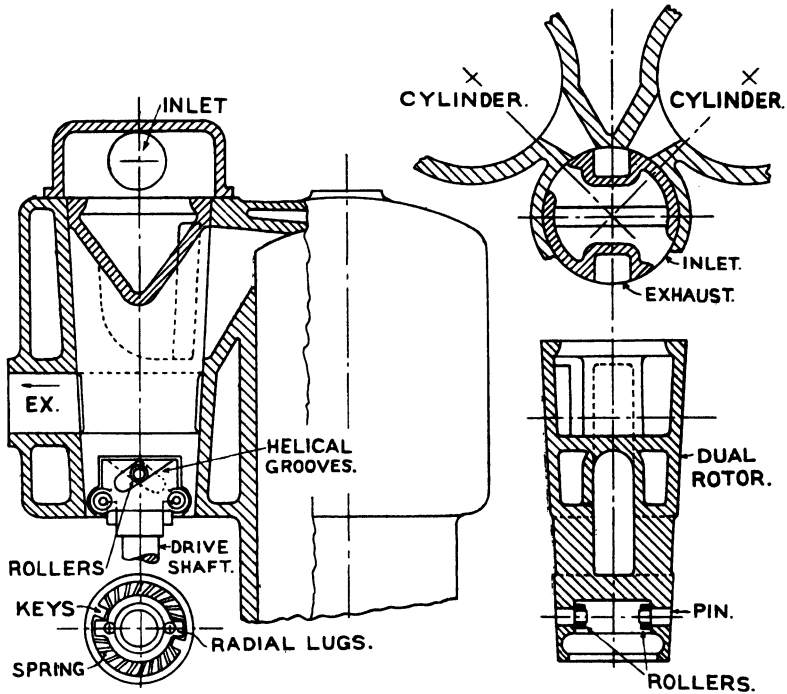



Fig. 47.
Russel dual valve arrangement. Single rotor feeding and exhausting two cylinders in turn.

because it illustrates the principle of employing a single rotor to feed two cylinders in turn and consequently making use of each port in the rotor twice in one revolution. The taper form of the rotor enables it to be ground to a perfect mating fit, but there is no guarantee that this fit will be maintained after the housing and valve have become heated up. The truth of the fit, following such irregular expansion as may take place on a casting of the complexity shown, is likely to vary considerably. A taper-fitting component is at all times sensitive to sticking, and although this device is carefully designed to avoid this tendency, the required strength of the spring in conjunction with the form of the helical

grooves is likely to be an extremely critical combination. No data are available as to the scheme of lubrication, but, for the above and other reasons, the arrangement cannot be considered fully to meet the problems involved in the production of a satisfactory design of rotary valve. If a dual type of valve were made to operate in a parallel bore with some form of independent sealing device between each of the cylinder ports and the rotor, the design would be on a much sounder basis and the friction could be greatly reduced.

It is an astounding fact that many clever engineers will go to a tremendous amount of trouble, and exercise remarkable ingenuity in a design, to overcome an adverse characteristic which may occur in operation, rather than expend the energy in producing a design in which the occurrence of such a characteristic is impossible. The design of the Russel rotary valve appears to fall within this class.

A. Darracq et Cie. Rotary Valve

A. Darracq et Cie., of Billancourt, France, were one of the earliest constructors of automobiles, and their designs, although somewhat crude in the early days, were always robust and could be relied upon to stand up to an enormous amount of hard work with infrequent trouble, so that when an engine with a rotary valve ("Sans Soupapes" is the term which the French give to every engine which has other than poppet-valves) was marketed and exhibited at Olympia by this company in the year 1911 it was fully expected that the innovation in design would stand up to the test of time. Unfortunately, following a very short period of manufacture and after only a few engines had actually been placed in the hands of users, the design was withdrawn from the market and receded into obscurity.

The engine as produced had four cylinders of 95 mm. bore with a stroke of 140 mm. Fig. 48 shows the general arrangement of the rotary valve and the position it occupies in relation to the cylinder head. It will be understood that the rotor, which turns at half engine speed, extends the full length of the cylinder block, parallel to the crankshaft, and that the valve is placed to one side. It is of unusual form compared with the basic styles of cylindrical rotors shown in previous illustrations. The novelty lies in a scoop-out (one for each cylinder) from the side of the solid rotor, and this cut-away provides a passage for both the inlet and exhaust, each in turn. It will be realized that although this feature provides a very simple arrangement, there are two fundamental defects in operation. At every change-over from exhaust to admission the scooped-out space of the rotor carries a charge of exhaust gases

which intermingles with the fresh charge on the ensuing suction stroke, and at the end of the admission stroke an equivalent volume of fresh carburetted mixture is carried round to the exit port and expelled into the exhaust pipe at the next opening of the exhaust.

The first defect may to some extent be reduced by employing a suitable timing, that is to say the momentum of the exhaust

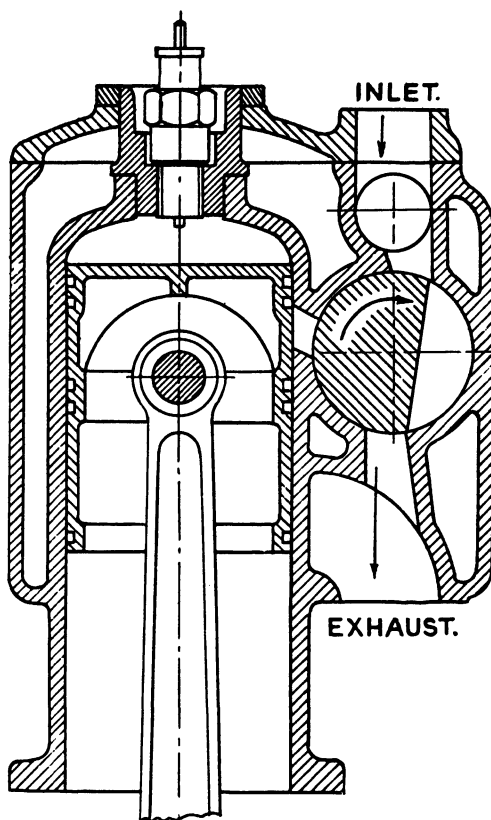



FIG. 48.
Darracq four-cylinder automobile engine with rotary valve shielded from the peak explosion pressure.

gases may be used to draw in some fresh gas at the time the change-over of the opening from exhaust to inlet takes place. The second defect is positive and can only result in a direct loss of useful combustible mixture, with a consequent increase in fuel consumption.

The question of timing, however, is complicated by the unusual position of the port leading from the cylinder to the rotor. A leading feature in the design of the engine, and one which it was hoped would bring success, was the masking of the valve by the

piston. That is to say, at the top of the stroke the piston entirely covers the port, so that at the moment of explosion the valve is completely shielded from the effects of combustion. Actually the valve is masked by the piston during one-seventh of the stroke, and thus provision is made for a transient seal at the end of the compression stroke and at the beginning of the firing stroke, when sealing is of most importance.

In this respect there is no doubt that the arrangement is of value, but the retention within the cylinder of such a large proportion of the products of combustion can only lead to low volumetric efficiency. As far as is known, no other sealing device was introduced into the design to ensure gas tightness around the port orifice.

The usual troubles of heavy friction over the whole surface of the valve, augmented by the pressure within the cylinder tending to force the rotor away from the port, the omission of a well-thought-out scheme of lubrication and the effects of temperature distortion, would all lead to early wear and leakage.

One or all of the above defects probably forced the manufacturer to discontinue production of the engine. Had the rotor been mounted on end bearings and fitted with some form of local sealing device of limited area at the port opening, then the arrangement might have progressed beyond the initial stage, in spite of the other shortcomings in the way of low efficiency and high fuel consumption.

The McGee Hemispherical Rotary Valve

Messrs. J. & G. McGee, of Glasgow, about the year 1911, produced an engine with a rotary valve which incorporated a feature of balanced pressure,* to reduce the working load on the surface of the rotor during the compression and firing strokes of the cycle. Fig. 49 is a sectional sketch of the cylinder head of the engine fitted with the McGee valve. It will be seen that the rotor is neither flat, conical, nor cylindrical, but hemispherical. The valve is constructed with double ports for both inlet and exhaust, and this feature of design enables a very liberal area to be given for the flow of the gases with comparatively small openings and ducts. This arrangement permits the valve to be made smaller in diameter but leads to duplicate pipes for the intake and exhaust, and adds to the difficulties of designing suitable manifolds.

The valve is driven by a vertical spindle carrying a spur gear pinion meshing with a gear wheel of twice the pitch diameter, so that in this design the valve turns at half the speed of the engine,

* Vide *Book of the Motor Car*, vol. 2. Rankin Kennedy, C.E. Caxton Publishing Co., Ltd., London.

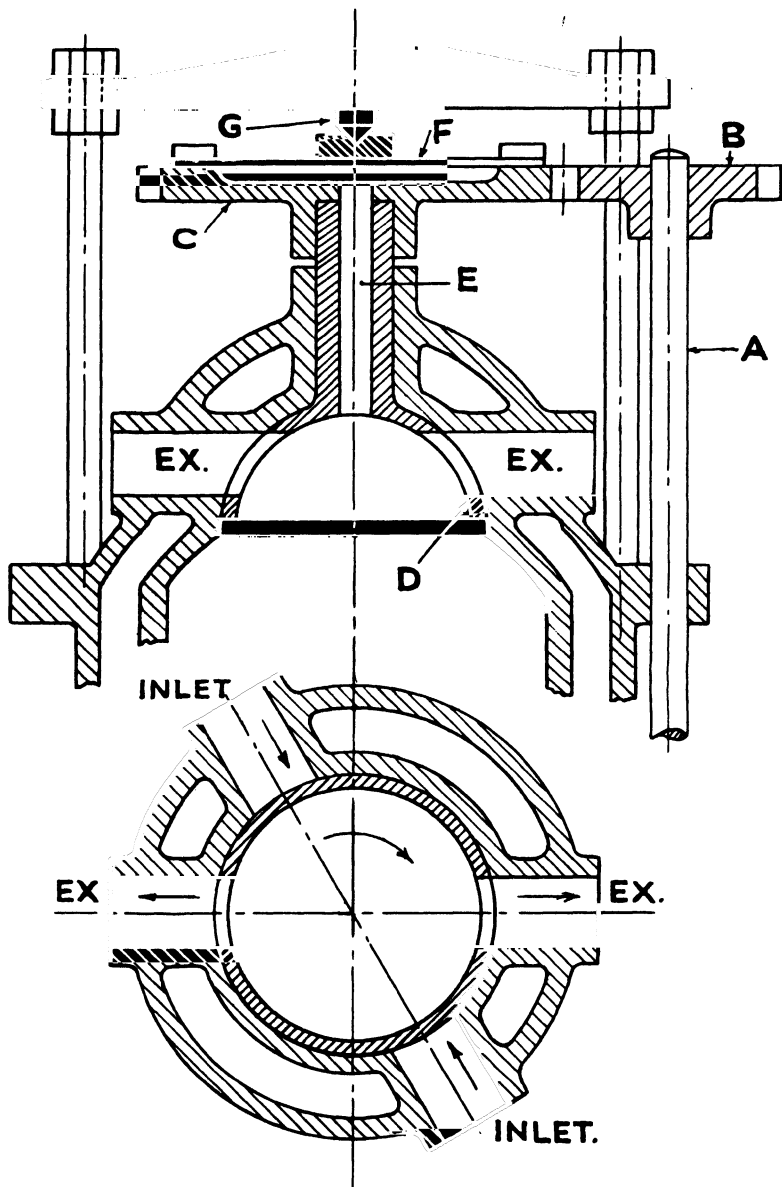



FIG. 49.
McGee hemi-spherical rotary valve with balanced pressure.

A—Vertical spindle. B—Spur-gear pinion. C—Gear wheel. D—Rotor. E—Drilled passage. F—Diaphragm. G—Anti-friction bearing.

in spite of the two openings in the rotor. The self-balancing characteristic is effected by the provision of a small drilled passage right through the valve spindle. This passage communicates with a shallow chamber formed on the upper side of the gear wheel. The chamber is covered by a diaphragm carrying at its centre an anti-friction bearing. The result is that any pressure tending to force the valve on to its seat is also communicated to the diaphragm, and by reaction tends to lift the valve to an equal extent. A spring, not shown, is of course necessary to maintain a nominal pressure between the surfaces and to prevent the valve being drawn away from the spherical head during the suction stroke.

The lubricating arrangements are not indicated, but an experimental single-cylinder car engine is reported to have been made which worked well and silently, and showed practically no signs of wear after lengthy service. Since the shape of the rotor is hemispherical, it cannot be considered an ideal machining job from the standpoint of general production, but it is possible that this particular contour may possess some intrinsic value not readily assessed on technical grounds. The double-ported valve has merit from a viewpoint of kinetic balance not usual in half engine-speed rotors. However, the really interesting innovation is the way that the feature of automatic control is introduced. Here, then, is a principle of considerable value, and one which should not be lightly dismissed. Other designers have used the principle of automatic pressure balancing, but it has generally been effected by somewhat different means and technique. The present design provides control with great simplicity and without a multiplicity of mechanical details.

The Itala Automobiles With Rotary Valve Engines

The reputation gained by the Itala Fabrica di Automobile of Turin for superfine manufacture and design has been a household word in the motor-car world for over a generation, and the standard of reliability achieved for their products in the mountainous country of Italy, if equalled by others at the period, was excelled by none.

Their models, always the result of strenuous road tests, terrific Alpine climbs of many miles in length, never failed to give the user a supreme satisfaction coveted by all contemporary manufacturers. It is therefore safe to say that when the Itala Co. embarked on a new programme of engines with a novel form of valve they were themselves thoroughly satisfied with the performance before attempting to place such an extraordinary departure from normal design in the hands of their customers.

It was in the year 1911 at the Olympia Motor Car Show that the first Itala models with a rotary-valve engine were announced to the British public, and although only one size of car was exhibited, namely the 35 h.p., two other models were on the way, the 25 h.p. with a bore of 90 mm. and 130 mm. stroke, and the 50 h.p. with a bore of 127 mm. and 150 mm. stroke. The smaller model was already going through its final tests in Italy and was reputed to be running with more than ordinary smoothness and power.

The 35 h.p. model exhibited had a bore of 105 mm. and a stroke of 150 mm. It had been exhaustively tested in experimental form before being standardized; in fact, the first car was driven a distance of some 30,000 kilometres, much of it on mountain roads, and for long periods at very high speeds, both on the level and on lower gears when climbing.

The rotary valve was well thought out in every detail and beautifully constructed. The speed of rotation was one-quarter engine speed, the rotors being placed vertically, one between each pair of cylinders, and serving each in turn, i.e. only two rotors for four cylinders. Each cylinder was provided with one port, the same opening serving for both exhaust and inlet, but the rotor was divided internally, as shown diagrammatically in Figs. 50, 51, so that the exhaust passed through the top and the live mixture from the bottom of the valve.

The features which distinguish this rotary valve from others are firstly the double porting referred to and secondly the system of water cooling. Both these features are intimately connected. The double porting makes possible a rotor speed of one-quarter engine speed, and though this does not reduce surface speed or wear, because it necessitates a corresponding increase in diameter, it has an important bearing on the scheme of internal water cooling. The enlarged diameter permits water passages into and through the body of sufficient area to allow enough water to be circulated to carry off all surplus heat. This alone would justify a quarter-speed arrangement, but another advantage accrues. As the valve rotates, the contained water is subjected to centrifugal action, and at a sufficiently high speed it would, if unrestrained, be nearly all thrown out or at least its circulation would be impeded, hence the great advantage of the low speed of rotation.

Fig. 50 shows how the water enters by a central port at the apex of the valve, passes down one side, across near the base and up the other side, to leave by an annular port also at the top. The circulation was made effective by the employment of a centrifugal pump impelling the water at a sufficient pressure completely to follow up the flow of the liquid due to centrifugal action.

Particular emphasis is laid on the fact that a valve internally cooled running in a chamber externally cooled was expected at all

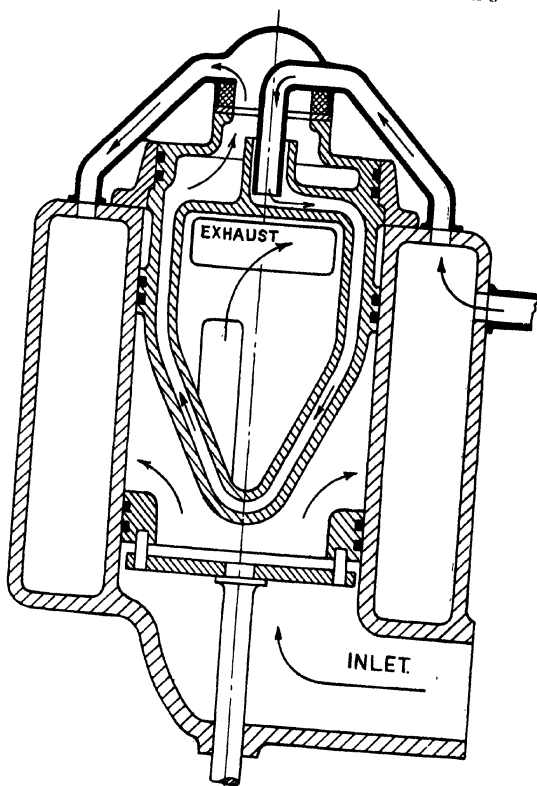



Fig. 50.
Itala water-cooled dual rotary valve.

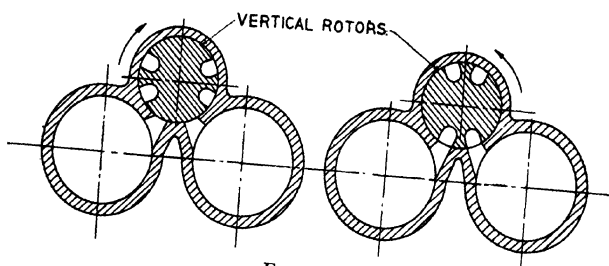



Fig. 51.
Layout of Itala four-cylinder engine with two dual vertical rotary valves.

times to maintain substantially the same character of fit, and that if it was a good fit when cold it would be equally so when hot. The feature was, of course, exclusive to the Itala at the period.

In this valve, with the induction below and the exhaust above, two piston rings are shown fitted to cut off any leakage that might take place from the cylinder along either path during compression and firing. These were stated to be effective and free from objection, because the rings in these circumstances do not rotate as the valve grooves slide over them. The valve body, none the less, was a good running fit in the casing; how good it is not possible to say. Excessive friction due to thrust from cylinder pressures was prevented by a clever and remarkably simple arrangement, which exactly counterbalanced pressure loads on the valve during the firing stroke. A small hole drilled through the valve body, in such a position as to be open to the cylinder when ignition takes place, carried pressure to the opposite side of the valve, and, by means of a recessed area in the chamber wall equal to the cross-sectional area of the port, resulted in the total transverse loads on the rotor being balanced. It will readily be understood that as no leakage takes place through the drilled hole from the recess, the arrangement results in no loss of efficiency, and further, as the contained volume is insignificant there is no measurable loss of compression.

The crankshaft carries a single helical or skew wheel flexibly mounted in such a way as to serve as a vibration damper. This meshes with a similar wheel on a side shaft which has two high efficiency worms, one right-handed, the other left-handed, each driving a wheel on the two vertical valve spindles, which, rotating in opposite directions, are therefore without resultant end thrust.

The valve spindles are connected through an Oldham-type coupling, designed as a breaking piece, so that if by any remote chance the valve should ever tend to seize, the damage will be confined to a part easily replaced. The provision of a specific breaking piece is no evidence of weakness, and it does not necessarily augur frequent trouble, but it is evidence of thorough foresight and is sound engineering practice.

Very little consideration of the Itala valve system tells one that the product was a superb piece of engineering and that it must necessarily be very silent. On the other hand, there may be a first impression that the slow-running valves would make for a sluggish engine and poor performance. These points are best disposed of by quoting from records of a run taken by the representatives of *The Autocar*,* who had the opportunity of driving one of the first cars to reach this country and who have given the results of their trial in the following words:

> "It was obvious almost immediately after the start that there was no ground for the fear that the engine would be sluggish. It would therefore appear that if the working drawings

* Vide *The Autocar*, January 6, 1912.

could be studied and the valve openings calculated, it would be found that they are at least equal to those given by large poppet-valves of full lift.

"During our test the roads were in a very heavy condition indeed, and we occupied ourselves largely in somewhat unkind acceleration tests; that is to say, we took a forty-mile run in hilly country in which right-angled corners at the bottom of the hills abounded. First of all we took Stoneleigh Hill, Warwickshire, which is so beloved of chassis testers that at times the police have to 'move them on'. This is a one-in-nine grade—actual, not estimated—and is approached by a rectangular turn. We took the turn slower than was necessary and, despite that fact, climbed the whole length of the hill on top gear. After that we attacked all sorts of similarly steep places equally badly situated for a start but the gradients of which have not been surveyed, though they are well known to all who test cars in the vicinity of Coventry as providing critical tests of the acceleration powers of a car if they are fairly approached; that is to say, if one turns the corners properly there is very little distance in which any speed can be regained, so that it is in the few vital strokes between the turn and the ascent that the question of making a clear top-gear climb or changing down is settled. At higher speeds we found the acceleration just as good, and while it is only reasonable to assume that this is very largely due to the carburettor, it follows equally that, however good the carburettor might be, it could not make up for any serious stricture in the induction system; indeed, it is not going too far to say that we absolutely convinced ourselves that the rotary-valved Itala was not only lively but one of the liveliest cars we have ever tried. At the same time, while its acceleration was so good, it was not uncomfortable or brutal, and, considering the length of the stroke, it is somewhat astonishing that one could take such liberties with the throttle without any discomfort. To the motorist who appreciates the running of a car we need only say it was a pleasure to drive, and we are able to add that liveliness is also one of its strong points."

The above record of a test carried out over thirty years ago cannot be considered otherwise than satisfactory, and the question may well be asked, how is it that an engineering achievement of such merit, with all the backing, experience and engineering ability of an organization such as the Itala Fabrica di Automobile, failed to retain a market?

The answer may well be found not on strict engineering premises but in the inability to command a sufficiently large market to enable economical production to be exploited. It was said at the time

that the Itala rotary-valve engine would never form part of a cheap car, and it is evident from a careful study of the design that it calls for the highest grade of materials and workmanship; in fact, every technical defect which might be possible appears to have been overcome by lavish expenditure and detail embellishment. More than this is required to ensure a successful commercial product, and these opinions are expressed: (1) there are too many parts, and (2) skilled experience is required for maintenance, greater than can generally be commanded from the ordinary repair organization in this country. The question of ready maintenance must be duly studied in connection with any new product if it is to succeed commercially as well as from an engineering viewpoint. Expensive items of construction must be avoided for successful marketing, even if technically meritorious performance is present.

Author's Early Proposal for a Quarter-Engine-Speed Rotary Valve with Straight-Through Ports

An arrangement of cylinder head for a four-cylinder engine is shown in Fig. 52, and this indicates one of the author's early proposals for which a provisional patent was filed covering certain principles in design. The First World War intervened and other activities prevented completion of the experimental engine. This in some ways was fortunate, because the design as it stands, without some obvious modifications, could not have been a complete success. The absence of any form of sealing device is now considered to be sufficient reason to condemn the conception. However, there are several features and ideas in the proposal worth discussing.

Firstly, it will be noted that the rotor runs the full length of the engine and is mounted on ball bearings at each end. A minimum clearance for lubrication between the rotor and the housing was provided, controlled by the differential thermal expansion of the two metals, cast iron for the rotor and aluminium alloy for the cylinder head. It was assumed that the temperature rise of the rotor would on the average approximate two and a quarter times the rise in temperature of the water-cooled aluminium head, and in consequence the annular clearance would remain nearly constant and no metallic contact would result.

If this condition could be attained, the only friction present would be that due to oil drag, as the thrust from the compression and explosion pressures is all taken on the ball bearings, which for all practical purposes are frictionless. From this fact it will be realized that the power-losses in a design incorporating this feature are extremely small, and no heat is added to the rotor or housing other than by the normal thermo-dynamic cycle of operations.

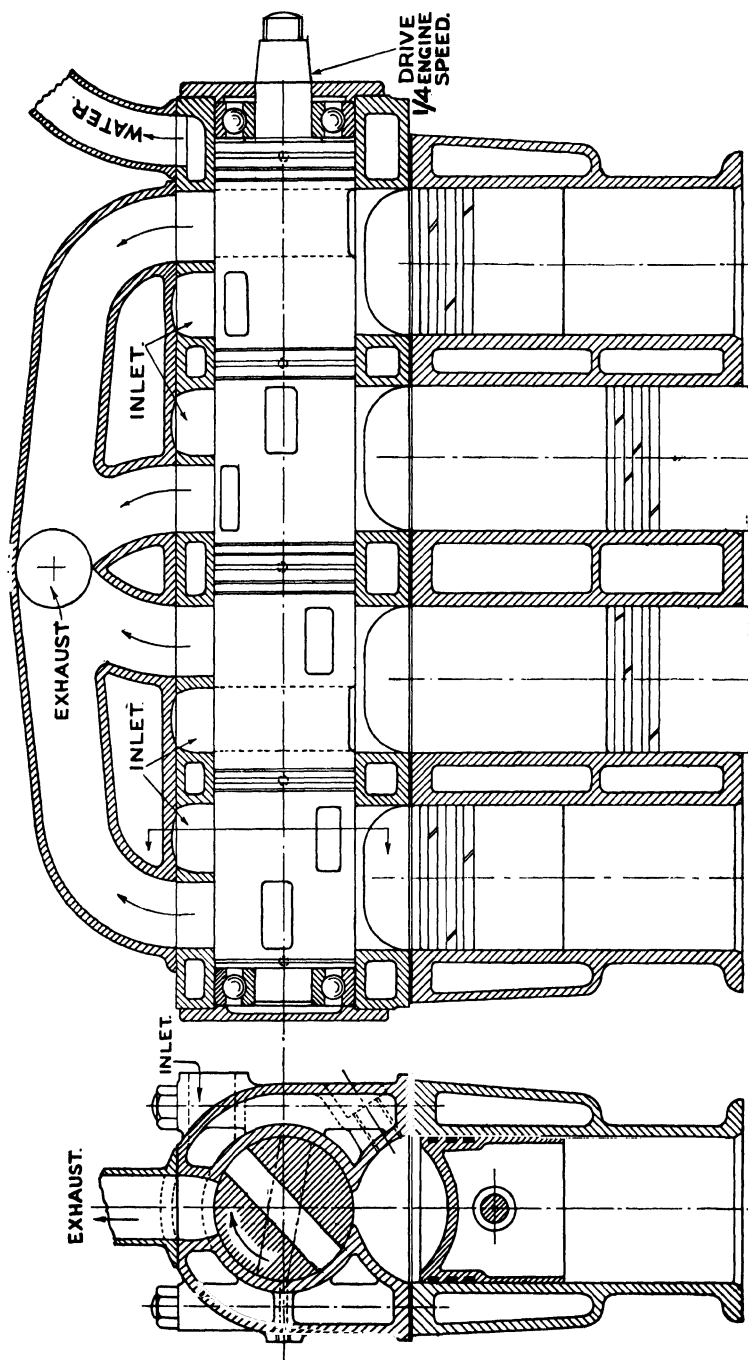



FIG. 52.
Quarter-engine-speed rotary valve with side-by-side straight-through ports.

Secondly, it will be noticed that both the inlet and exhaust ports in the rotor pass right through in the form of slots, and these are spaced side by side in correct angular relation to each other to suit the timing specification. This port arrangement lends itself to excellent streamlined paths for the flow of the gases, although a handicap is introduced in the way of limited breathing areas.

The straight-through port requires a quarter-engine-speed for the rotor, and the illustration is a good example to emphasize the comparatively large diameter of rotor necessary to obtain even a moderate valve-piston-area ratio, which in this case works out at 0·13 for an engine of 70 mm. bore. This is admittedly poor, and is due to the side-by-side disposition of the ports, but there are compensating advantages in the improved streamline characteristics. The general construction allows liberal water-jackets, free circulation and a simple exit to the radiator.

Not the least interesting point in the design of this proposed engine is the feature of an annular clearance between the rotor and the bore of the housing maintained by the end bearings. This principle is a meritorious one but of doubtful service to the cause of the rotary valve, without the addition of an efficient seal. The advantages from a power-loss point of view should, however, not be lost sight of.

Early Work of Mr. R. C. Cross

Experimental work on rotary valves applied to a motor-cycle and other I.C. engines was commenced by Mr. R. C. Cross as early as 1922, and it can be said that since that date he has designed and tested more types of engines employing the rotary-valve feature than any other constructor in this country, or, indeed, of any other country.

Many of these early engines have produced remarkable performances, and much detail information on his early designs and practical experience has been made available in a paper read by him before the Institution of Automobile Engineers.* Some of the results obtained indicate that compression pressures as high as 260 to 390 lb. per sq. in. can be used with fuels of 66 octane, and with this same fuel over 70 h.p. per litre and 156 lb. b.m.e.p. has been gained with smooth flexible operation. These are truly remarkable figures for that period. In Mr. Cross's experience the substitution of the cool rotary valve for the red-hot poppet-valve (this was before the general use of sodium-filled valves) permitted compressions to be raised by several ratios over what would be permissible for a poppet-valve engine unless special fuels were used.

Research work carried out in order to study the influence of

* Vide *Proc. of Automobile Engineers*, 1935.

charge turbulence, and on special shapes of piston and combustion chambers calculated to give extreme turbulence, indicated that in the cool combustion chamber of a rotary-valve engine the valve

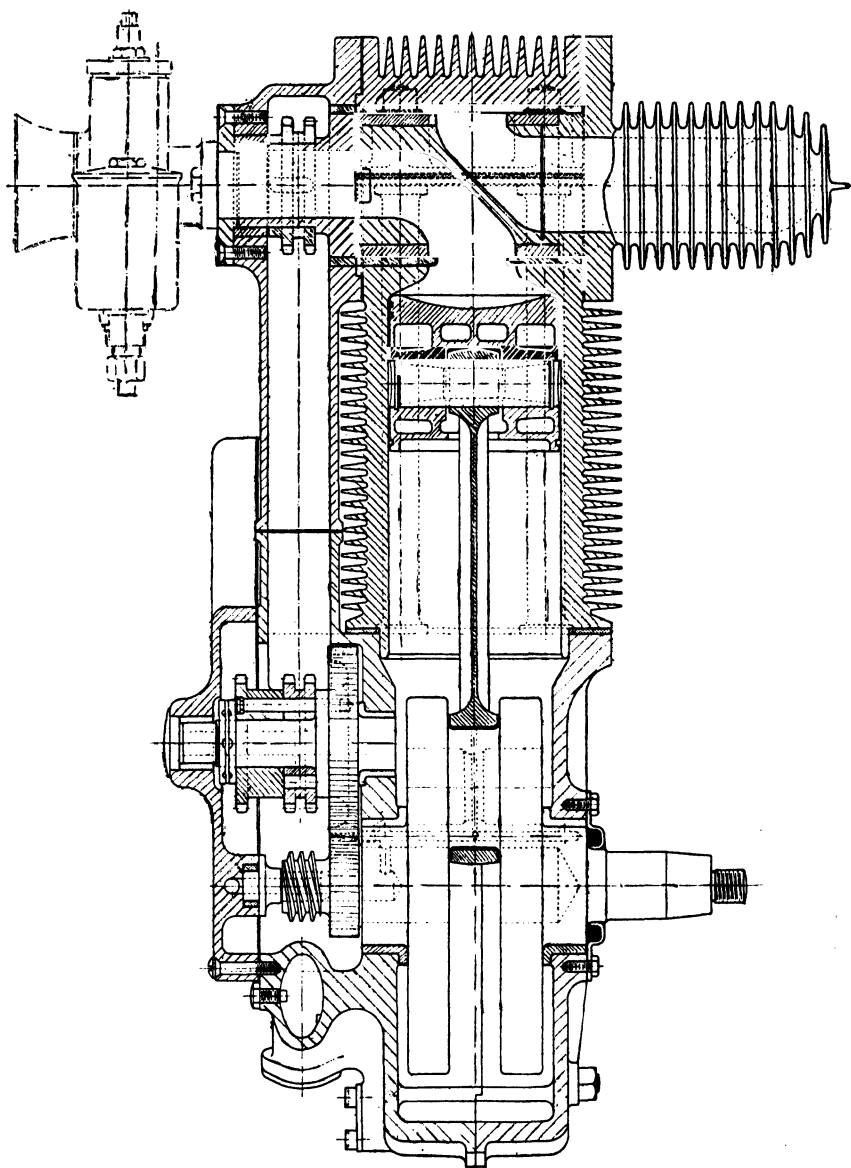



FIG. 53.
Cross's early single-cylinder motor-cycle engine with chain-driven cylindrical rotary valve parallel to crankshaft.

of turbulence did not appear to be pronounced to any great extent. In the poppet-valve engine the case is known to be different, because turbulence prevents charge stagnation adjacent to the hot exhaust valve.

VALVE FORMS. Several experimental types and sizes of engines had been produced in the period covered by the paper, i.e. from 1922 to 1935, some with liquid-cooled rotors and others in which the rotor was cooled only by contact with its surroundings. The advantage of an uncooled valve was stated to be a quicker rise in temperature when starting from cold, thereby tending to prevent deposition of petrol. The uncooled valve is simpler and less expensive to produce and it was found to be perfectly satisfactory for small engines and proved capable of giving high performance. Large engines were made with hollow rotors, liquid cooled, and these gave a lower general temperature, thereby helping the lubrication problem.

MOTOR-CYCLE ENGINES. One of Cross's early single-cylinder motor-cycle engines is shown in Fig. 53. The rotor is placed parallel to the crankshaft and is chain driven. This engine had an aluminium valve with a nitri-cast iron or nitralloy steel shell, and the sprocket centre formed the induction pipe. It is of the straight inlet type, the carburettor being on the axial centre line of the valve. The system has the merit of great simplicity and gives great power, but it has the disadvantage that the carburettor is in a rather unprotected position, and is therefore susceptible to injury.

An alternative arrangement was made in which the valve was driven by a spindle carried on bearings outside the induction pipe, with the carburettor opening at right angles to the axis of the valve, the spindle passing through a close-fitting hole in the pipe and engaging with dogs in the valve. In this design the rotor is fully floating and is cast in one piece in nitri-cast iron. It did not, however, give quite the power of the type shown in Fig. 53, because of the higher resistance of this form of induction pipe. Since the carburettor is in a warmer situation and more protected, the arrangement in practice was found to be rather more flexible. Another engine, shown in Fig. 54, has the valve driven by way of a vertical shaft with two pairs of bevel gears. The upper gears provide the two-to-one ratio, and the lower gears being of equal size the rotor runs at one-half engine speed.

FOUR-CYLINDER ENGINE. An early design of four-cylinder Cross engine with the valves all in line proved very successful. The valve in this case was designed to run at only one-quarter engine speed. The rotor was made up of two pairs of valves joined resiliently in the middle and driven from one end. The arrangement provided for the cooling water to pass right through the valve. In the middle of each pair of valves is an angular groove, which

registers with the inlet pipe and communicates with the cylinder port by way of a channel running at right angles to the groove. This channel is correctly placed so that it registers with the cylinder at the correct period to give the desired timing. The exhaust

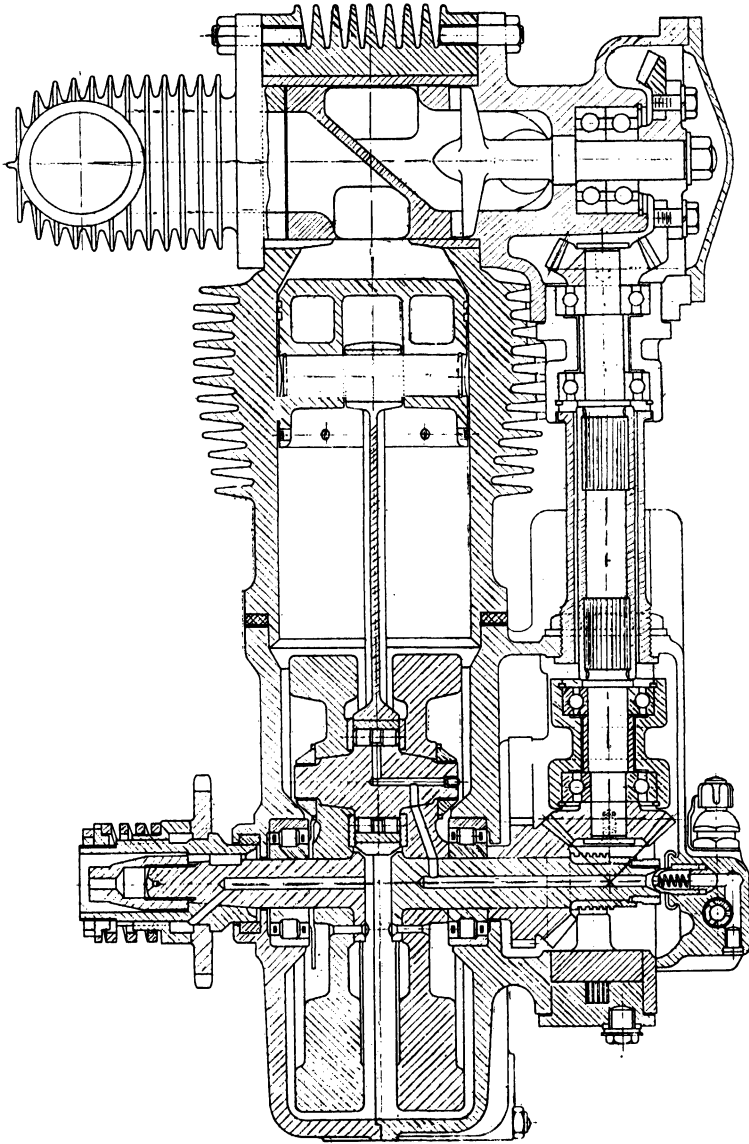



Fig. 54.
Cross motor-cycle engine with rotary valve driven by a vertical spindle and bevel gears.

leaves at each end of the head. This style of valve provides a fairly smooth outlet, but the quarter-engine-speed rotor is large in diameter and the weight greater than that of the half-engine speed type. The design, however, has the advantage of more space for ports and cooling media.

A half-engine-speed type of rotor is shown in Fig. 55, and there the arrangement of the ports is entirely different, the exhaust ports passing diagonally across the valve to register with other ports in the valve housing. It should be noticed that the cylinder head is supported on side plates and independent of the cylinders.

SEALING DEVICE. The design of the sealing device employed by Cross at the period is similar to the basic type shown at C, Fig. 12, and it was used in all the above engines. The design is based on the following conclusions:

(*a*) That the sealing line of contact should be as near to the port edge as possible in order to reduce the length of seal contact to the minimum, and also to reduce surface exposed to flame.

(*b*) That the sealing port edge must be relatively resilient in order to maintain contact with the valve.

(*c*) That the sealing edge should be pressed more firmly into contact by the gas pressure, but should be gastight, without the assistance of gas pressure.

(*d*) That the sealing edge should be provided with some means of cooling.

The value of sealing at the extreme inside edge of the cylinder-head port is considered by Cross to be of great importance, a small gas leakage being a serious disadvantage, as it tends to heat up both the valve and the sealing device, and also tends to destroy the oil film. It is further highly desirable that the sealing edge of the port should have a slight degree of resilience so that it is able to follow the surface of the rotor under all the conditions of temperature expansions and distortions that are ever likely to occur in operation. Mr. Cross emphasized that a carefully designed engine, and particularly in the type having a liquid-cooled valve and housing, needed a very small degree of resilience.

After much experimental work he established the design and methods of manufacture for what may be termed the "lip" type of sealing device already referred to, which consists of making the port edge of the correct thickness so that it will deflect a few thousandths of an inch without stressing the metal very much. In manufacture the edge of the port was raised by a press-tool operation for a width of approximately 3 mm., and then machined off true to the valve bore, but so that the edge stood above the surface of the valve bore by only a few thousandths of an inch, the amount of "lift"

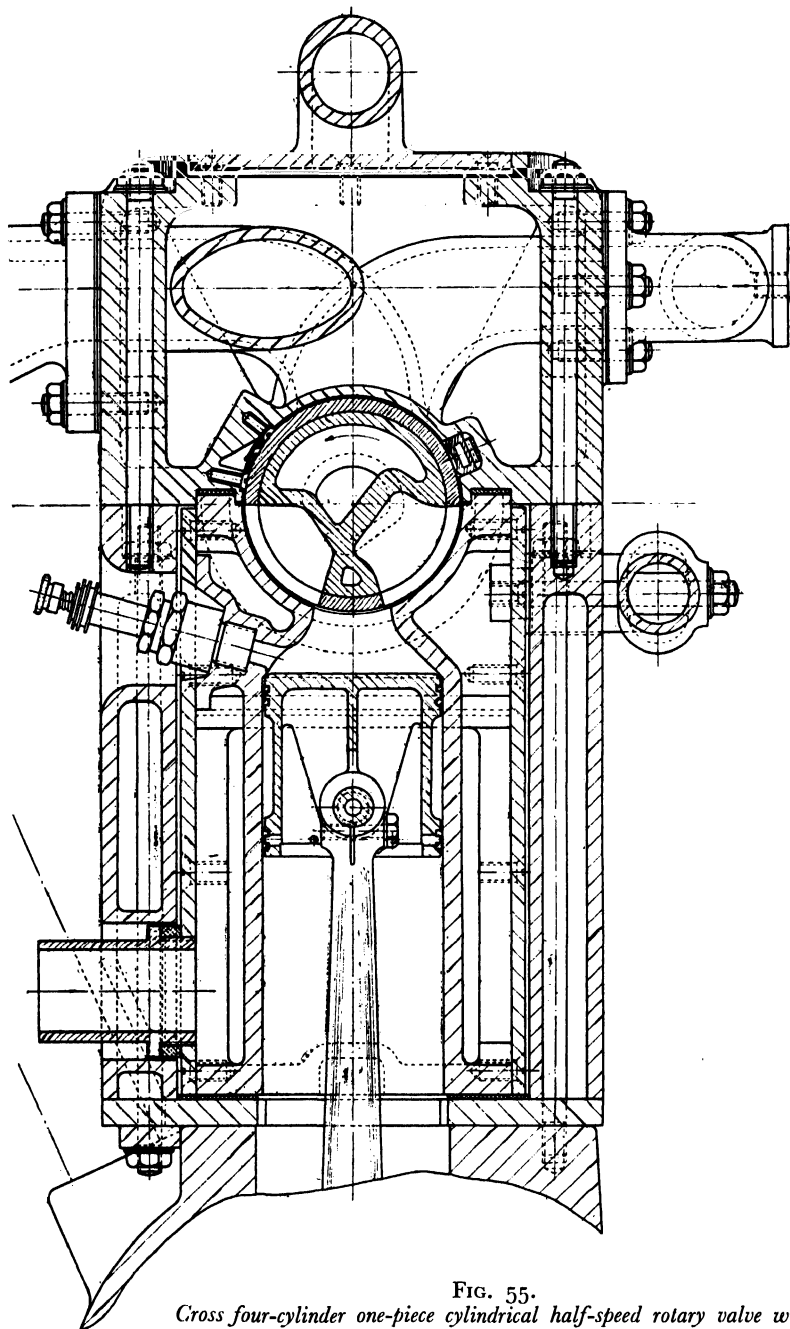



FIG. 55.
Cross four-cylinder one-piece cylindrical half-speed rotary valve with head supported on side plates.

depending upon the conditions existing, but usually being not less than 0·0015 in. and not more than 0·006 in. When the valve is assembled in the bore the edge comes in contact with the valve face, with the result that the metal around the port edge is deflected and the lip is resiliently held against the valve surface. This artifice resulted in a good seal, and the higher the gas pressure in the cylinder the more firmly the edge of the port is pressed against the valve. As a matter of convenience, the port edge was usually formed in the valve bush, and in practice this gave good results over long periods of running. In order to keep the port edge as cool as possible, the inlet port in the valve is made about 8 to 10 mm. wider in a longitudinal sense than the port-hole in the cylinder head, while the exhaust port in the rotor is narrower than the cylinder-head port and gives the flexible edge a certain amount of screening during the exhaust stroke. The action of the inlet gases playing upon the edge of the lip during the induction stroke was reputed to produce a marked cooling effect.

In some cases the valve bush was made from a hollow bronze casting through which cooling liquid could be circulated, the chief object being to increase the resistance of the port edge to burning, and not to ensure a cool surface to flame in the combustion chamber.

It was also found that the conditions of sealing were greatly improved when means were provided whereby the clearance between the valve and its housing had automatic adjustment. This was carried out by splitting the valve housing on the horizontal centre line of the valve and at right angles to the axial centre line of the cylinders. Long bolts were then used to fasten the top cap to the crankcase, distance pieces being inserted accurately to space the cap from the crankcase. The distance pieces were in some cases tubular pillars round the centre of the bolt or alternatively were constructed in the form of stiff hollow side plates as shown in Fig. 55.

In all these designs the construction permits the cylinder to float axially between the crankcase and the valve itself, the cylinder being pressed towards the latter by resilient packing between the cylinder base flange and the crankcase. Soft packing also fills the gap between the bottom and top halves of the rotor housing. The full effect of this is that the cylinder is pressed against the valve and any clearance is automatically taken up. The gas pressure further tends to hold the cylinder more firmly against the rotor. This arrangement made things easier for the resilient lip of the cylinder head port to perform its sealing function, and also provided good thermal contact between the valve and the housing.

LUBRICATION. Cross was one of the first to appreciate and point out the difficulties which had to be overcome in effecting proper lubrication of a rotary valve.

Oil, he said, may be put on the valve in the smallest quantity that will provide satisfactory lubrication and not be recovered, or it can be applied more copiously by means of a circulatory system in which the oil is fed in on one side of the valve and removed from the other side by a scraper device. Cross developed a circulatory system of lubrication which gave most promising results, and at the same time a low oil consumption and smokeless exhaust. The scheme is shown in Fig. 56.

It was found, after many practical experiments, that the best place to apply lubricant to the surface of a rotary valve is about the lateral centre line. This means that in all practical designs the oil feed must come over the middle of the inlet and exhaust ports in the rotor. This creates a varying resistance to the oil feed, inasmuch as the feed is subject to suction when the inlet port passes

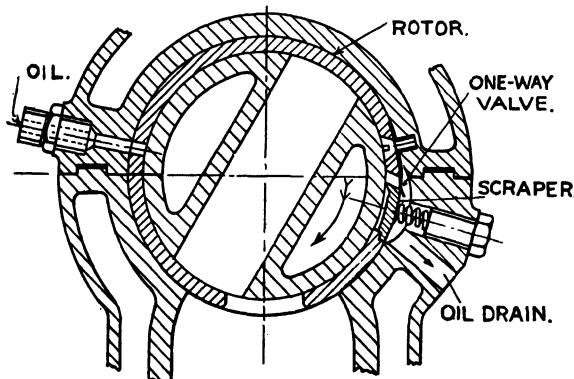



FIG. 56.
Cross oiling system with detail of scraper and non-return valve.

the oil feed and to pressure when the exhaust port passes it. It will be realized that as the throttle is opened the inlet suction falls off and the exhaust pressure increases. This state of affairs is exactly the opposite of that which would be desired, inasmuch as when the throttle opening is small, and both the heat to be dealt with and the valve loading are slight, resistance to the oil feed is changed to a state which assists oil to enter the valve chest. Added to this, the scraping device (described in detail later) which removes oil from the face of the valve naturally finds it more difficult to carry out this function and pass the oil back to the source of supply, which is under atmospheric pressure while the valve chest itself is under partial vacuum. Therefore the obvious course of procedure is to control the supply of oil so that as the throttle closes the supply is reduced, and when the throttle is open the supply is increased.

This may be done in several ways. If the oil is fed by a positive-action pump, as in the case of motor-cycle engines, then a pump in

which it is possible to vary the output by means of a control coupled up to the carburettor control can be used. An alternative method for motor-cycle engines, which was used with some degree of success, is to fit a pump which is capable of delivering a much larger quantity of oil than is normally necessary and to fit a small pressure-regulating device operated by the pressure in the valve chest. This opens and closes a jet to the rotor, allowing more or less oil to pass as is required and the remainder of the oil is by-passed against a spring-loaded piston valve to the source of supply. In the case of multi-cylinder engines, where large quantities of oil can be circulated from a gear pump, it was found satisfactory to fit a regulating device to control the pressure to the whole engine, in place of the conventional spring-loaded pressure-regulating valve. A gallery pipe along the valve chest is provided with a series of jets, one to each valve. The size of these jets is, of course, a matter of considerable importance, and generally has to be established by means of trial and error.

THE OIL SCRAPER. The scraper blade attacks the surface of the rotor and scrapes the oil from it into a recess at the back of the blade. To cope with the pressure fluctuations in the valve chest an arrangement has to be made to fit the scraper with a one-way valve which permits oil and some slight exhaust gas with it to pass the valve, but prevents any oil from returning when the inlet port passes the scraper blade. The design of this one-way valve is of great importance, probably much more important than the design of the scraper blade itself. Fig. 56 shows an early design of a one-way valve acting directly on the scraper blade. This valve can with advantage be supplemented by a second non-return valve in the drain pipe away from the cylinder head.

Even when the engine is running at very low speeds there is some pressure in the exhaust system, and although inlet suction is much greater than the exhaust pressure, yet the scraper is made to work effectively, since neither gas nor air can return backwards from the scraper to the valve, because of the non-return element. This prevents the gases from carrying the oil backwards, but when the exhaust port in the valve passes the scraper the pressure from it opens the non-return valve and disperses what small quantity of oil has accumulated during one revolution of the valve.

When the throttle is wide open the scraping becomes much easier, since the suction is small and the pressure in the exhaust system is considerable.

By the careful study of the oil supply to the valve and the scraping of the valve surface it has been found possible to lubricate the valve effectively so that it will withstand any condition to which it is subjected, and at the same time give a reasonably low oil consumption, and also to prevent the oil from interfering with the

mixture, which would be detrimental to carburation and ignition. The scraper used in present-day designs is of very much more robust construction than that shown here, but the principle on which it works is the same.

THERMAL EXPANSION AND DISTORTION. Cross very early found that the rotary-valve engine lent itself to aluminium cylinder-head construction, and he soon realized that the high rate of expansion of this material did not introduce so many problems as might at first be expected. In the air-cooled engine with contact-cooled valve, the temperature of the valve is naturally higher than its surroundings, and if the whole valve, or at least its shell, is made from ferrous material, the difference in temperature compensates for the difference in expansion.

Although the rotors in the Cross design did not depend entirely upon the excellence of fit in the valve housing, yet he stressed the opinion that the smaller the clearance the better, within certain limits, as demands upon the sealing device were then less exacting, and there was less possibility of any exhaust gas passing down the side of the valve and so diluting the incoming charge. This is a possibility against which there must always be a safeguard. The liquid-cooled engine with liquid-cooled valves offers some advantages, as liquid cooling not only reduces the temperature of the parts, and thereby decreases expansion, but also tends to maintain a more uniform temperature which reduces distortion to negligible proportions. Even in the simple air-cooled engines, however, the use of the split housing takes care of the clearance and very little distortion results.

In some designs, although a liquid-cooled cylinder and head may be used, the valve is only contact cooled. Good results were obtained with this type, but the system does not give the best control of expansion. The valve temperature varies considerably with variations of load, while the cylinder and head temperature varies but little once the engine is warmed up. Before the introduction of the split head, some air-cooled engines showed stiffness when starting from all cold, although the water-cooled types with contact-cooled rotor were very free. With the split-head type the resistance to turning when cold is about equal to that of the conventional poppet-valve engine.

PERFORMANCE. All the engines described above were stated to be smooth running, and would not readily detonate. The mechanical silence was remarkable, and the exhaust note not unpleasant. Flexibility was above the average, together with a good combination of low-end torque and high-speed characteristics.

The performance on low-octane fuel was excellent. A b.m.e.p. of 157 lb. per sq. in. has been maintained at 4400 r.p.m. in one engine and 154 lb. per sq. in. at 5700 r.p.m. in another. The

second value represents 68 h.p. per litre, and the same engine developed 70 h.p. per litre at 6100 r.p.m. using petrol of octane number 66.

A motor-cycle fitted with a 250 c.c. engine with flexible type carburettor and efficient silencer was proved to be capable of maintaining a road speed of 80 m.p.h. using the same grade of fuel.

A long-distance test on a 350 c.c. air-cooled engine gave an oil consumption of 4100 m.p.g. for the whole engine, and this figure was improved on a water-cooled model.

The actual cylinder barrel and head temperatures were observed to be lower than in a normal poppet-valve engine. A test carried out on a motor-cycle engine after the machine had completed a fast run terminating in a steep hill climb showed the maximum cylinder-head temperature to be only 129°C.

The foregoing information shows that even at that period Mr. Cross had a most profound knowledge of the characteristics of the rotary valve as applied to internal combustion engines, and that he had covered a great amount of ground towards the development of a successful engine. But the arrangements described and used for effective sealing indicate that this function was secured by utilizing several principles in unison. Firstly, the liner surrounding the rotor was made with as close a fit as was possible and the bush and housing were split to allow for expansion. Secondly, the edges round the port were deformed and rebored to ensure greater local contact near the opening, thus providing the essential lip-seal. Thirdly, the cylinder, in one piece with the head and the lower half of the valve chest, was elastically mounted, in order that the pressure within the cylinder could exert the total load from the explosion on the bottom half of the bush which constituted the actual seal guarding the explosion pressure. The area of the cylinder bore multiplied by the working pressure gives a figure representing the possible load on the under-side of the rotor. The top side of the rotor carries a similar load due to reaction, therefore doubling the forces of friction.

Although this combination was shown in practice to give workable results, it is fair comment to say that the scheme includes far too many variables, any one of which might make or mar the efficiency.

Since the bush and housing were split to allow the gap to open slightly with thermal expansion of the rotor, it is evident that under some conditions the full reaction to the thrust would be taken on the cap, hence the rotor would be fully loaded on both the top side as well as on the under-side. The power-losses in these circumstances would be considerable.

For the above reasons the arrangement could not be accepted as a full solution to the problem. The designer himself, indeed,

realized some of the shortcomings, and later designs indicate a departure, employing an entirely new principle of controlled loading which is described in a subsequent chapter covering design of modern engines.

Minerva-Bournonville Rotary-Valve Engine

The Minerva Co. of Antwerp was the first Continental firm to produce Knight's sleeve-valve engines exclusively, this policy being continued from 1920 to 1925, and when the company took up the development of a rotary-valve engine it did so in the critical attitude resulting from its long experience with sleeve-valve manufacture.

This novel engine, which came to be known as the Minerva-Bournonville,* was not the invention of the Minerva Company, but of M. Bournonville, a Belgian engineer residing in New York. After he had produced his first engine, with encouraging results, M. Bournonville realized that it needed developing, and admitted that the persons best fitted for this work were Knight engine specialists. The Minerva engineers, after experimenting on the bench and on the road for a period of two years, pronounced the Minerva-Bournonville rotary-valve engine developed to such an extent that it could challenge comparison with the best Knight engines, while justifying claims to cheapness, simplicity and reliability not to be found in any other type of engine with which they had had experience up to that time.

THE MECHANISM. The Minerva-Bournonville has a cylindrical rotary valve mounted on the cylinder head and running the length of the block. Pockets are cut on the rotor face by means of which communication is made between the cylinder and the intake and exhaust passages. For each cylinder there are three pockets on the rotor; thus the valve rotates at only one-sixth crankshaft speed. The rotation of the valve is in the opposite direction to the main shaft, and, as can be followed in the series of diagrams A, B, C and D, Fig. 57, the pocket which has assured communication with the exhaust port immediately afterwards provides for admission of a fresh charge as shown on diagram B and thereby obtains the benefit of the cool gases. The uncut portion of the cylindrical rotor next covers the port into the cylinder during both the compression and explosion strokes (diagram C); then the second pocket comes into play for the exhaust indicated in diagram D, followed on the next cycle by the third pocket.

This low rotational speed is a favourable feature, because, whilst giving a liberal port area, it allows time for the heated parts of the

* Described and illustrated in *The Autocar*, May 20, 1927.

rotor to cool off before again coming in contact with the incoming fresh charge. At an engine speed of 4000 r.p.m. the valve rotates at approximately 666 r.p.m. only, which is a very low speed in internal combustion engine practice. If this low speed had been the only good quality in the design of the rotary valve it is doubtful whether the Minerva-Bournonville engine would have been developed to the practical stage. However, there are several other principles in the assemblage which merit attention.

PREVENTION OF SEIZURE. The valve is carried in a special type of elastic bearing which makes it almost impossible for the

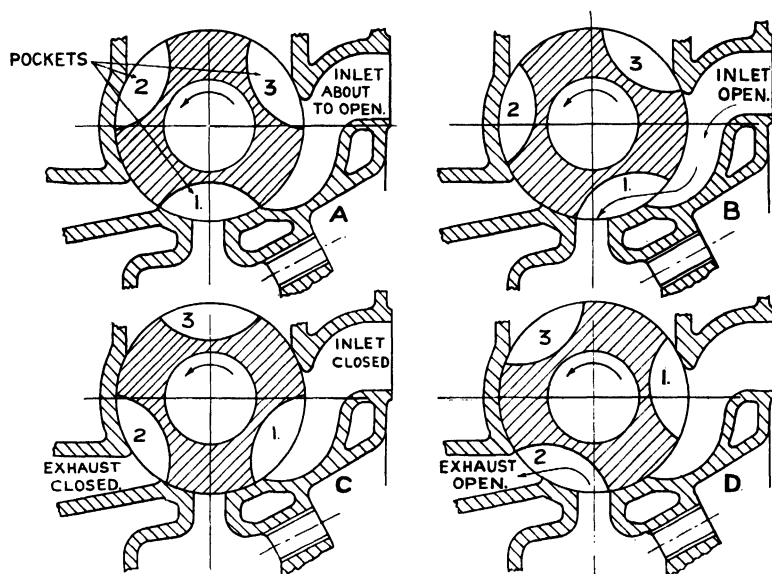



FIG. 57.
Diagram of Minerva-Bournonville one-sixth engine speed rotary valve in four positions. A, top dead centre. B, induction stroke. C, explosion stroke. D, exhaust stroke.

rotor to seize. This will be readily understood by referring to Fig. 58. The diameter of the boring along the cylinder head is slightly greater than that of the valve, the difference being shown somewhat exaggerated. This difference is calculated so that under conditions of maximum thermal expansion the rotor will be free to turn in the barrel. The barrel is not completely circular, the upper portion having vertical walls, and in this cavity is placed a semicircular cast-iron shoe surmounted by a cast-iron wedge. Between the shoe and the wedge there is a recessed steel ball; the upper surface of the wedge is in contact with an inclined surface in the cylinder head, and by means of spring-operated plungers placed horizontally in the cylinder casting pressure is exerted on the wedge, and through

it on to the shoe and the rotary valve. In combination with the plungers are screwed studs which are adjustable through calibrated springs to allow sufficient pressure to be exerted on the valve to prevent it lifting under the explosion. After the engine has been started up the studs are screwed in until there is no movement of the wedge. This movement can be detected with considerable accuracy by means of a pin going through the centre of the stud and touching the face of the wedge. When correct adjustment has been

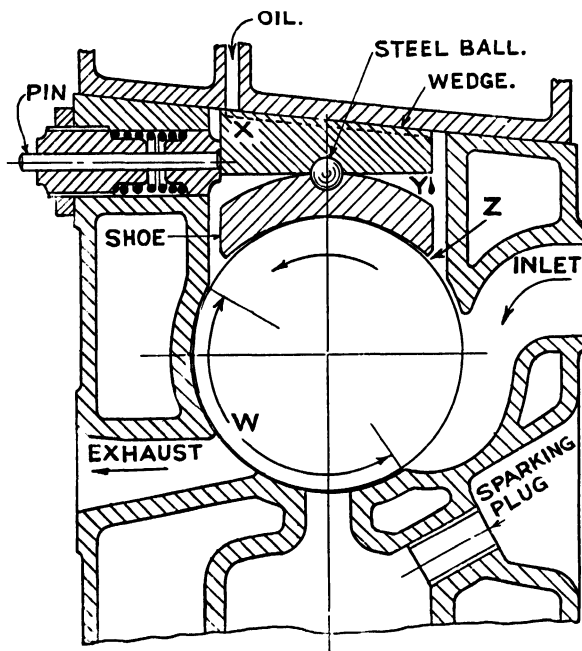



FIG. 58.
Cross section through the Minerva-Bournonville cylinder head, showing valve wedge-gear and lubrication.

attained, movement of the wedge can take place only for one of two reasons: dust or other foreign matter finding its way between the rotor and shoe, or increased friction by reason of temporary lack of lubrication.

In each case the rotor would tend to carry the shoe around with it, and by reason of this movement it would cause the wedge to move outwards, against the compression of the spring, thus automatically relieving the pressure on the rotor and preventing seizure. The steel ball interposed between the wedge and shoe is not normally under pressure, and its only purpose is to transmit any slight rotational movement of the shoe to the wedge.

For the sake of simplicity, a single rotor, shoe and wedge have been referred to. Actually, in the six-cylinder engine the rotor is in two parts, the shoes are in four parts—two parts for each half of the rotor—and there are two wedges. A steel ball is set in the centre of each shoe, and there are four studs and springs—two for each wedge. The shoes and the wedge are in the same grade of cast iron as the cylinder block, while the rotary valve is of a special grade of steel and case-hardened. The shoes are cut out of a tube and are bored slightly larger than the diameter of the rotor, the difference being about one-thousandth of the diameter of the valve. They are turned externally to a uniform thickness throughout their width. The wedges are irreversible and have an angle of 7° to 7½°.

As the part of the barrel marked W in Fig. 58 has a diameter slightly greater than that of the valve, when starting up from cold the edges of the pockets are the only parts which bear, but when a normal working temperature has been attained the valve maintains contact over the entire circumference.

Being in two parts, it is possible to drive the rotary valve from the centre. A spindle, the external diameter of which is smaller than the internal diameter of the rotary valve, has two forks at 180° in relation one to the other, mounted on the central portion of its length. One rotor is slipped over the shaft from each end and engages with the forks by means of slots cut in their extremities. The forward end of the drive spindle is rectangular and engages with a slot milled in the face of the driving pinion. The drive is taken from the crankshaft pinion to an intermediate pinion with a reduction of 3 to 1, and from the intermediate pinion to the rotary valve pinion there is a double roller chain drive with a further reduction of 2 to 1, thus giving a total reduction between crankshaft and valve spindle of 6 to 1.

LUBRICATION. While the special type of bearing makes it almost impossible for the valve mechanism to seize, the lubrication system is carefully carried out and will be easily understood by referring back to Fig. 58. From the rear end of the rotary-valve shaft an oil distributor sends one drop of oil per 150 revolutions of the engine through five leads connecting respectively with the intake manifold and with four points on the top of the wedges at X, near the spring-controlled studs. Each wedge has a deep oil groove along its entire length and criss-cross grooves by means of which the lubricant is distributed to the thin end. Here the oil drops down on to the shoe at Y and thence on to the rotary valve. The direction of rotation of the valve causes the oil to be carried round between valve and shoe, entering the fine clearance at Z.

A good feature of this valve mechanism, from a maintenance standpoint, is the ease with which it can be dismounted, for it is possible to withdraw the whole of the mechanism in a fraction

of the time necessary to take out a set of poppet-valves. Further, no special tools are required for this work.

On the rear end of the cylinder block, covering the valve boring, is a cast-iron plate carrying the oil distributor and held in place by studs and nuts. When this plate has been removed, when two screws have been taken out and one oil-connection has been broken, the rotor, the shoes and the wedges can be withdrawn by hand. On the bench this removal can be carried out in less than one minute, and on the road the time need not be more than five minutes if an opening has been cut in the dashboard for this purpose. The rotor, if properly adjusted, is unlikely to require attention; nevertheless, accessibility must always be accounted a feature of merit in any design.

Road and Bench Tests. Much experimental work was carried out on two types of engine: a four-cylinder model of 80 × 112 mm. bore and stroke which, having big ports and, for that period, the high compression ratio of 5·91 to 1, developed 60 h.p. at 4000 r.p.m., and a more normal six-cylinder type, with a compression ratio of 5·2, developed 34 h.p. at 3000 r.p.m.

In addition to the bench tests, one of the engines was kept constantly on the road for eight consecutive days and nights, in the hands of different drivers, and during practically the whole of this time 4000 r.p.m. were maintained. In addition, the engine was run for a whole day immediately behind another car on a very dusty road, in order to ascertain what effect, if any, dust would have on the valve mechanism.

It was stated by the Minerva engineers that it was practically impossible to make the valve seize even under the most clumsy handling. Obviously, it is possible to create such conditions that some part of the mechanism must suffer, but in all such cases it was found that the pistons, the big ends, or the main bearings seized before defects developed in the rotary valve. The engine under review was conventional in all features other than the valve mechanism, and was built with a fixed cylinder head, but it would have been quite possible to have adapted the valve mechanism to an engine with a detachable head. The Minerva-Bournonville engine was extremely silent, and had the quality of maintaining its silence throughout its life.

Reports say that the engine had a degree of flexibility above that usually associated with internal combustion engines of the period. Petrol consumption figures showed it to be equal to that of a normal type engine, and its oil consumption rather less than that of a sleeve-valve engine.

It will be gathered from the above review that before marketing this engine all steps appear to have been taken that might have been likely to make it a success. The design is a straightforward

manufacturing proposition, and certainly the components are less difficult to make than those in a sleeve-valve engine, in the production of which none had more experience than the Minerva engineers and none were in a better position to assess the standard of service demanded by the public. In spite of all this, the product failed to secure any outstanding success.

A minor criticism can be laid against the design, inasmuch as if the engine was rotated in the reverse direction to normal, as under conditions of backfiring, the rotor would tend to draw the wedge inwards and in consequence might tighten up the whole valve; it is not known whether this defect gave any trouble in practice. The prime necessity for correct adjustment of the springs on the wedge may have been too critical a job in the hands of others except the actual makers. The device in any circumstances must have been somewhat sensitive. In fact, the design is dependent upon an adjustment which controls the amount of friction and from this aspect the design is intrinsically indeterminate. There is no seal, other than the pinching action between the lower boring and the shoe, therefore the friction and heat to be dissipated are comparatively high, even though the pressure on the wedge is adjusted to a nicety.

Mr. F. M. Aspin's Experimental Work With Rotary Valves

The early experimental engines produced by Mr. F. M. Aspin, and the numerous changes in design leading up to his successful

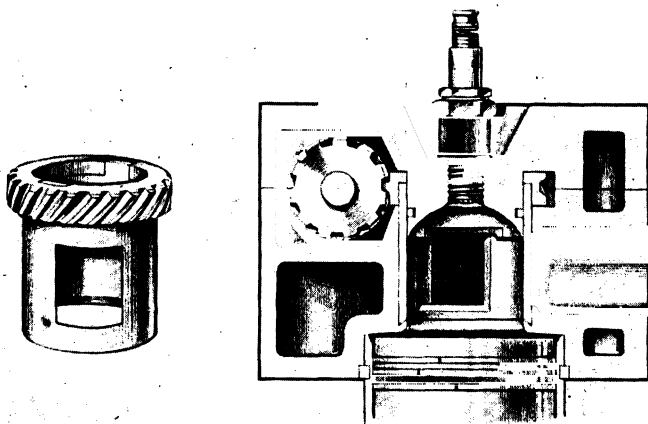



FIG. 59.
First type Aspin engine made. Rotating sleeve with port. Rotation half engine speed.

prototype engine with the conical rotating combustion chamber, are best studied by examining a series of sketches.

EARLY DESIGNS. Fig. 59 shows the first engine made and the rotor in this case is a thin sleeve of cylindrical form, rotating at half engine speed. Sealing at the bottom end is effected by a close-fitting bevel edge, whilst the top of the sleeve is provided with an ordinary compression ring. Gas tightness around the port orifice depends on quality of fit. It should be noted that the ignition plug is situated in the centre of the head. A limiting factor in the design is the port size, and the cylindrical rotor was therefore abandoned in favour of a solid rotating member of conical form provided with an eccentrically disposed combustion space, as indicated in Fig. 60,

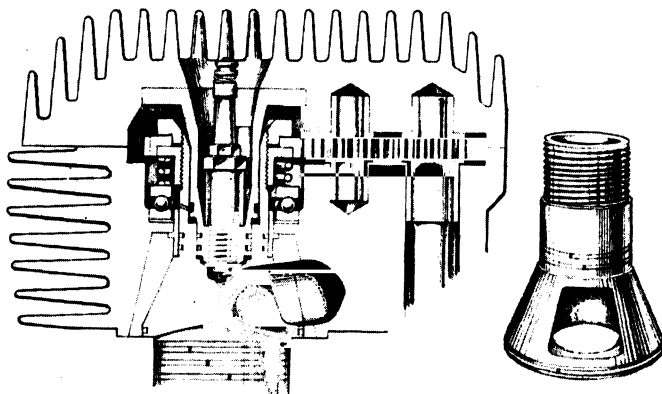



FIG. 60.
Second Aspin engine made. Eccentrically disposed combustion space. Sparking plug central.

permitting an increase in the area of the port. It will be seen that the sparking plug is again disposed in a central position. The rotor was made in phosphor bronze running in a cast-steel insert, the latter being a tight fit in the aluminium head. The performance of this engine was stated to be gratifying, but the oil consumption high. The major trouble experienced with this design was the high temperature attained by the sparking plug, which is uncomfortably shrouded from the atmosphere and is unsuitably placed to allow the heat to be efficiently carried away by conduction.

The next step in design was an attempt to avoid the hot plug. This is indicated in Fig. 61, where the rotor is driven from the underside instead of from above. The change introduced mechanical difficulties of considerable magnitude, but these were all overcome by an ingenious form of drive. The rotor unit is made from case-

hardened nickel steel, running in an aluminium-bronze housing, the latter again being fitted tightly into the cast-aluminium head. The sparking plug in this design is in a much better position for the ready transfer of heat by conduction, but rather at a disadvantage for proper cooling of the cylinder barrel. The performance, however, was stated to be very good, and a considerable mileage was run with some satisfaction. The major defect of the arrangement is recorded as seizing of the rotor if given full load without thoroughly warming up the whole engine.

At this stage in the experiments an important modification in design of far-reaching effect was introduced, a feature that carries right through to the present productions and without which it is highly probable success would never have been achieved. The location

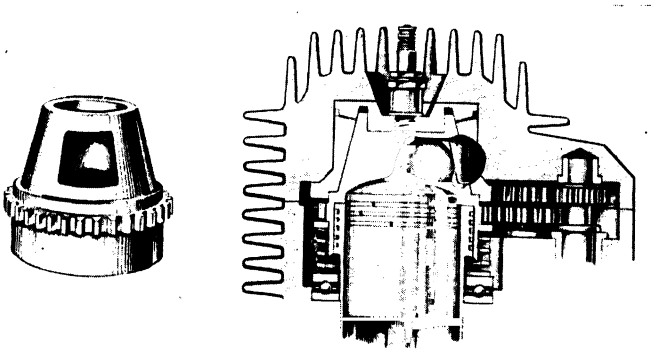



FIG. 61.
Aspin experimental engine with conical rotary valve driven from the underside to obtain a cooler running plug.

of the sparking plug was changed from the central position in the head to the side of the engine, but this was more than a change of situation—it introduced an entirely new principle in the fact that the sparking-plug points are only open to the combustion space and exposed to the hot gases for a very short interval of time, just sufficient to start ignition at or near top-dead-centre; the rotor then covers the aperture to the plug points until the next charge is ready for firing. The performance from this engine was extremely satisfactory, and it is well to take particular note of the materials used for construction. The rotor of solid duralumin has the surface sprayed with high carbon steel. This working surface, of course highly finished, runs in a duralumin bush fitted into the usual cast-aluminium head. From this it will be realized that the whole assembly is manufactured from materials with characteristics of comparatively high thermal conductivity and with approximately

similar coefficients of thermal expansion, but—and this is important—the mating surfaces where friction occurs are of dissimilar composition, namely hard carbon steel on duralumin, a combination which has been found satisfactory in other applications. The cylinder capacity of the experimental engine was 249 c.c. and a maximum speed of 14,700 r.p.m. was attained. Consumption of lubricating oil was fairly high and the engine generally required further mechanical and detail development.

The first design of a multi-cylinder liquid-cooled engine is

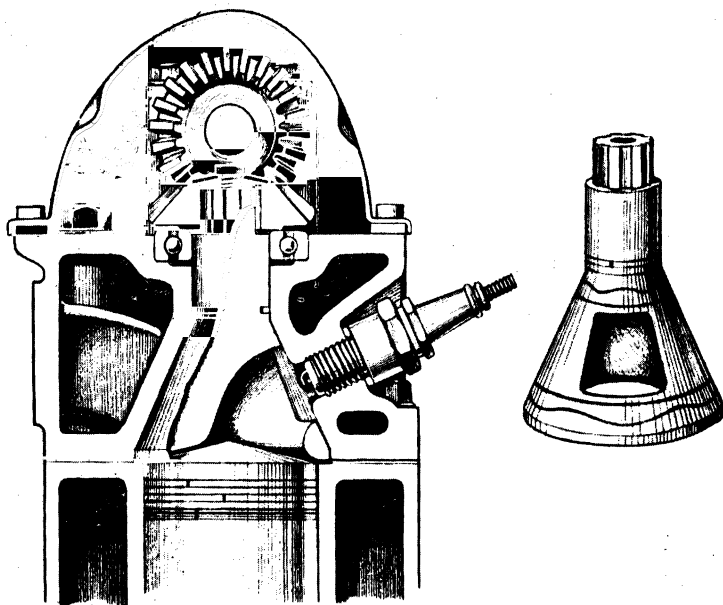



Fig. 62.
First Aspin multi-cylinder liquid-cooled engine.

shown in Fig. 62. The general features of the system are similar to those used in the previous experimental air-cooled unit. The materials, however, are different. The head is of cast iron with the rotor of the same metal but tin-plated on the outside working face. This combination of mating friction surfaces had to be abandoned on account of seizure if worked too hard before the lubricating oil had thoroughly warmed up. Special notice should be taken of the wavy grooves on the conical rotor that are provided for the purpose of spreading the lubricant, and the annular ball bearing on the stem of the rotor for taking the side thrust from the driving gear, a most important mechanical requirement if loads,

other than sealing, are to be segregated from the working surface of the conical rotor.

THE FORCES ACTING ON A CONE. At this juncture it is important and of value to notice that in all the above arrangements of valves the full pressure in the cylinder acting on the projected area of the rotor is transmitted to the working surface to promote the seal. Furthermore, since the form of the valve is conical, the load is considerably augmented before reaching the working surface, varying with the design angle of the taper. It can be shown that:

$$\text{Load on surface of rotor} = \frac{\text{load on base}}{\text{sine of the angle of cone to the axis.}}$$

Example:

Diameter of cone at base .. = 3 in.
Area of base .. .. .. = 7·06 sq. in.
Average explosion pressure .. = 150 lb. per sq. in.
Angle of cone to axis.. .. = 25°
Sine of 25° .. .. .. = 0·422

Then,

$$\text{Load on surface of rotor} = \frac{7{\cdot}06 \times 150}{0{\cdot}422} = 2017 \text{ lb.}$$

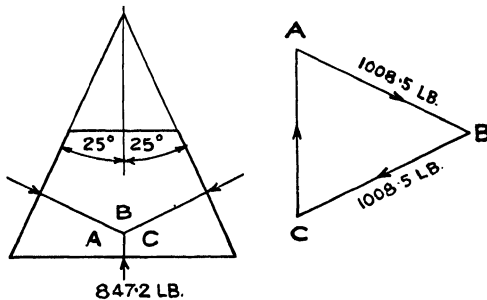



FIG. 63.
Diagram of forces on a conical rotor.

Fig. 63 shows the diagram of forces. The resulting friction and the power-loss are high, although the unit load per square inch of bearing surface may be moderate and quite in keeping with general practice and permissible from the standpoint of efficient lubrication.

PROTOTYPES. Proceeding to the next stepping-stone in the Aspin

developments reference should be made to Fig. 64, wherein the problem of friction has been attacked by a method of construction quite unusual in previous designs, and with somewhat astounding technical results.

In some respects this engine may be considered to be the prototype leading up to the design on which the more recent productions are based. Descriptive articles covering the performance of this engine first appeared in the automobile journals in 1937. At that date Aspin had established and tested two sizes of engine—a 67 mm. × 70·5 mm. single-cylinder air-cooled motor-cycle type

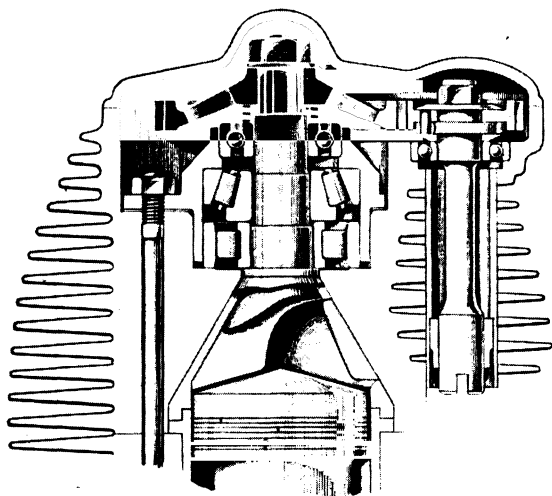



FIG. 64.
Sectional elevation of Aspin single-cylinder motor-cycle engine with taper thrust bearing to take the explosive load off the conical surface of the rotor.

unit and a light four-cylinder horizontally opposed aircraft engine of 83 mm. × 80 mm. bore and stroke, also air cooled.

The main points of interest in both these engines lie in the combustion head and the mounting of the rotary valve, and as the principles of construction are the same for both engines the following description is confined to the single-cylinder motor-cycle type unit.

The whole engine was, not without good reasons, designed with a view to meeting excessively high pressures and revolutions. This implies the use of very stiff reciprocating parts, short stroke in relation to the bore, and nitralloy journals on the crankshaft. The piston of robust construction was made in "Y" alloy; the connecting rods were of ceralumin—an aluminium alloy of 30 tons tensile strength.

The two illustrations, Fig. 64 and Fig. 65, show respectively a sectional elevation of the upper part of the cylinder and a diagrammatic cross section through the head in plan, looking upwards.

The solid rotor functions as an internal rotary head for both inlet and exhaust and as a heat shield to mask off all hot areas from the burning charge. Following the design features of previous experiments, the rotor is conical in general form, with a shaft extending vertically upwards from its apex to carry a Timkin thrust bearing, on which the main axial load is taken, while ball bearings are included for centring purposes. The shaft is extended upwards

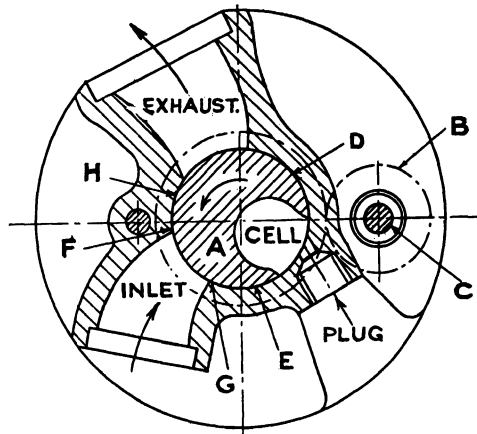



FIG. 65.
Diagrammatic cross section of Aspin cylinder head looking upwards. A, rotor. B, engine-speed pinion. C, driving spindle. D and E, sealing surfaces. F and G, port edges. H, wall between inlet and exhaust.

and carries the half-speed-driven gear wheel actuated by the engine-speed pinion which surmounts the vertical driving spindle.

When the valve is first assembled the outer face of the rotor, which is coned to 60°, has a definite metal-to-metal contact with the housing and takes all the load, but it is run-in to a condition just short of metallic clearance, so as to admit of a sealing oil film, which is fed in by a small upwardly inclined channel (not shown in the illustration) and distributed round by cut grooves on the outer cone face of the rotating head.

A vital point in the general scheme is the eccentrically placed "cell", as it is termed, which breaks through the outer periphery of the cone and is seen in Fig. 64, lining up with the exhaust port, and in line with the sparking plug in Fig. 65. The sponsors prefer to regard this cell as a kind of rotating combustion chamber, in

the orbital sense, and in this respect it differs fundamentally from the ordinary basic type of hollow conical rotor.

A little consideration will show that the compression ratio is determined by the cubic capacity of the cell, plus the clearance over the piston, related to the swept volume of the cylinder, and that the valve timing is defined by the position of the several port edges.

The originator of this engine considers that the special feature of the rotating cell has considerable influence on the remarkable results of improved combustion without any signs of detonation.

Whatever the true causes for the outstanding results attained with this engine, there is no doubt of the excellence of combustion, nor of the very efficient energy conversion from latent fuel calorific

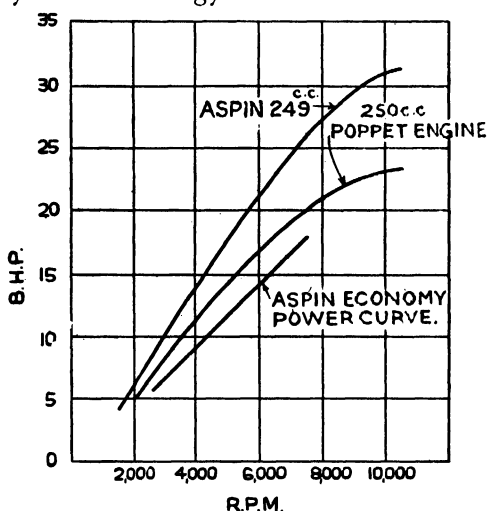



FIG. 66.
Power curve of Aspin 249 c.c. engine compared with that of a 250 c.c. engine with poppet-valves.

value into horse-power. This was demonstrated on the prototype by the fact that the exhaust gas at the exposed port during maximum output on test was so cool that the hand could be passed in front of and at three or four inches from the port without any undue discomfort. Another point is that no flame or luminosity is visible even in a dark room with the exhaust manifold off.

The horse-power curve of this remarkable engine is shown on the accompanying graph, Fig. 66, and is compared with the horse-power of a normal high-output engine of the period, of equal capacity but of poppet-valve type. The economy curve shown is also quite out of line with ordinary spark-ignition engine performance and more nearly approaches that expected from Diesel engines.

The actual engine from which these tests were taken was reported to have completed over 700 hours' running at high loads and speeds without any undue wear, and had sustained a full-throttle run at 14,000 r.p.m. for many hours without any apparent ill effects. The compression ratio was approximately 14 to 1, which at that time was considered unusually high.

Attention must again be drawn to two very important features in this prototype engine. First, the sparking plug is shrouded

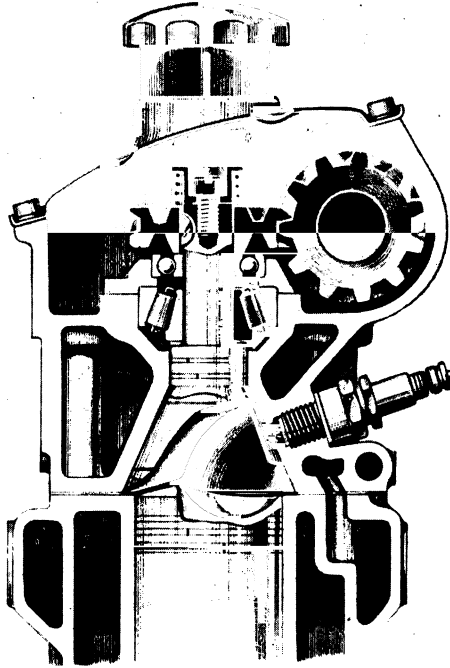



FIG. 67.
Aspin multi-cylinder water-cooled engine.

except during a very short period necessary to initiate combustion. Secondly, the rotor is mounted with a Timkin bearing so that all thrust is taken on this bearing and other loads are taken on separate bearings, and not on the cone seating. The valve is therefore not in metallic contact with the housing. Surface friction, except for oil-drag, is non-existent, and in consequence mechanical power-losses from any cause are almost zero. The principle of mounting the cone with a clearance is very important, but it is not easy to take advantage of in larger engines, where distortion and thermal expansion are greater.

It should be recorded that the cylinder head was of cast aluminium and the rotor of high tensile steel with aluminium internal filling, but it will be appreciated that where non-metallic contact is aimed at, the question of suitable mating is not of such vital importance as when actual rubbing takes place. The performance shows the almost phenomenal b.m.e.p. of 180 lb. per sq. in. The oil consumption was stated to be equal under full load to that of conventional air-cooled engines of the period but rather heavy under light loads.

Fig. 67 represents a multi-cylinder engine developed on the same lines as the air-cooled engine but provided with water cooling to both the cylinder and the head. In this case the rotor was of bronze running in combination with a cast-iron housing.

In a multi-cylinder engine the difficulties of the extremely fine adjustment required to taper bearings, or, alternatively, the running-in process to produce the same results, is not a practical undertaking. It is therefore interesting to note that the design with taper-thrust bearings has been abandoned, and in the modern Aspin productions adherence to the principle of actual clearance between rotor and stator is not in evidence.

CHAPTER VI

BURT MCCOLLUM SEMI-ROTARY SLEEVE-VALVE ENGINE

THE single sleeve valve with semi-rotary motion was patented in 1909 by the late Peter Burt, a Britisher, and the invention was adopted by the firm of Argylls, Alexandria, N.B. The patent covered an engine comprising a sleeve valve working "inside" the cylinder, with a novel mechanism for actuating the sleeve both with an oscillatory and a semi-rotary motion. McCollum was a Canadian who took out a patent for an engine wherein the sleeve encircled the "outside" of the cylinder. The Argyll Co., to cover their interests, purchased the invention. For this sole reason the two inventors' names are usually coupled. Engines, past and present, using the principle always follow the fundamentals of the Burt arrangement with the "inside" sleeve.

OPERATING MECHANISMS. Various methods adopted for operating the sleeve are given in Figs. 68, 69 and 70. That shown in Fig. 68 is the form employed in Argyll engines where a sliding

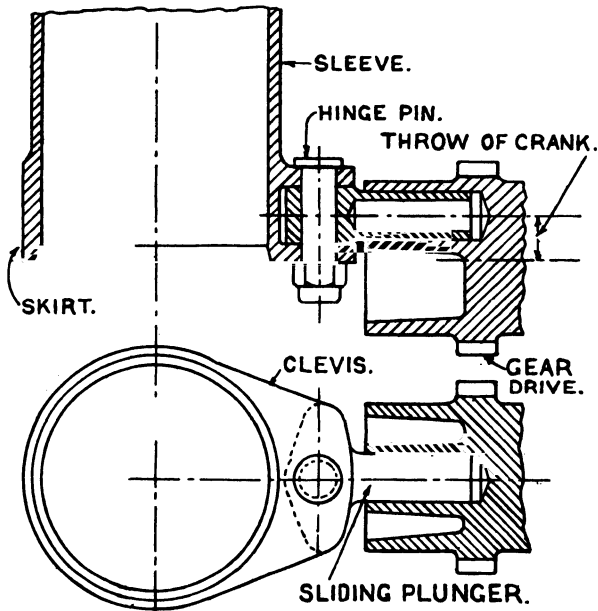



FIG. 68.
Operating mechanism for semi-rotary sleeve valve used on early Argyll engines, incorporating hinge pin and sliding plunger.

plunger in a crank arm is used in combination with a hinge pin to produce the oscillatory and semi-rotary motion. The sliding plunger works in a clevis formed integral with the skirt of the sleeve and therefore increases the effective weight of the sleeve assembly. In spite of this objection, a large number of engines were constructed with this type of operating mechanism and generally gave good service.

A more attractive form is that shown in Fig. 69, in which a ball-and-socket joint is used in place of the hinge pin. This arrangement is lighter, more compact, and the crank pin acts at a point nearer to the axis of the cylinder, resulting in a relatively greater angular movement to the sleeve. It is also the least expensive to produce.

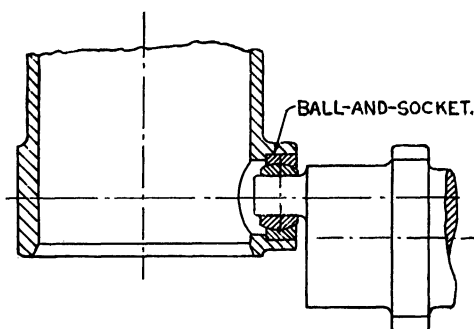



Fig. 69.
Alternative form of operating mechanism with ball-and-socket joint providing a relatively greater angular movement to the sleeve.

The operating mechanism shown diagrammatically in Fig. 70 is that used in the early Picard Pictet engines manufactured under Burt's patent. From a mechanical point of view the design is very sound but the cost of manufacture is greater. Two half-speed cranks are employed, one at either side of the cylinder, and the sleeve is operated from the centre of a coupling bar connecting the two crank pins. The sliding block at one end of the coupling bar allows for any small errors in synchronism of the two half-speed shafts and any slight discrepancies in the throws of the two cranks. The reciprocating parts have considerable mass and the design is unsuitable for high-speed engines.

The Sleeve. A sleeve operating with a semi-rotary motion is clearly an improvement over a plain revolving sleeve as the combined oscillatory and rotary movement prevents scoring of the cylinder barrel and the outside of the sleeve. It also assists the lubrication of the two surfaces. The chief fault of the double-

sleeve valve engine, as used by Knight, was the difficulty of disposing of the heat from the piston through two films of oil and two sleeves, and the friction loss is high. The Burt arrangement is superior in both respects.

A special mixture of ordinary cast iron and hematite iron was used by the Argyll Co. for the sleeve castings. The proportions of the mixture were critical and some trouble was experienced due

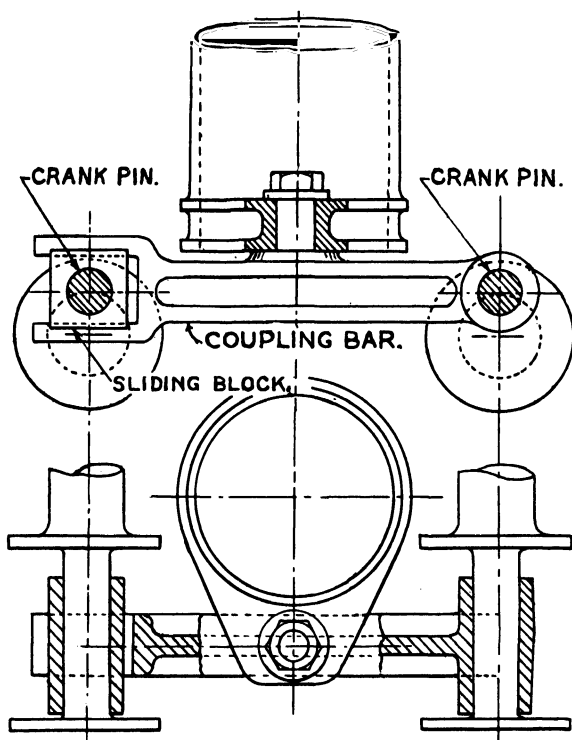



FIG. 70.
Operating mechanism for semi-rotary sleeve used on Picard Pictet engines.

to cracks developing in the metal between the port apertures, unless the constituents and pouring temperature were strictly controlled. A suitable thickness for the wall of the sleeve when cast in this mixture is from 0·1 to 0·125 in., depending on the size of the cylinder. The weight of the sleeve and skirt as a cast component is therefore considerable. Sleeves are now generally manufactured from steel.

The shape of the ports is important and it is necessary to adopt a special outline to gain the largest area of opening with a minimum

of crank throw. The ideal shape is shown at A, Fig. 71, but this is expensive to manufacture and the sharp corners are liable to start incipient cracks in the material. Outlines more suitable for production are those shown at B, C and D. The straight flank, port C, is a good manufacturing proposition, although slightly smaller in effective area for a given operating crank throw. That shown at D is in general use in modern engines.

ARRANGEMENT AND NUMBER OF PORTS. A variety of port

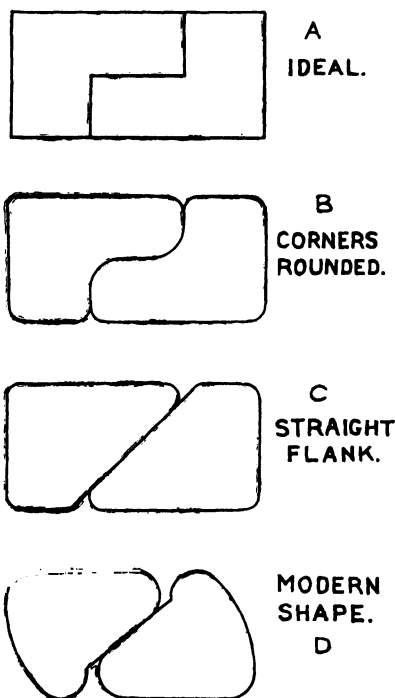



FIG. 71.
Typical port outlines.

arrangements may be employed. Several are shown in Fig. 72, which is approximately to scale and set out on a basis of the sleeve circumference. The greater the number of ports the smaller the necessary crank throw for a given area of valve opening, thereby reducing the overall dimensions of the driving mechanism and height of engine, but leading to complication and increasing the port machining time. The largest areas are gained by using the fewest number of ports, but with the disadvantage of a longer crank throw.

It will be noticed that at least one "dual-purpose port", that is one which serves alternately as inlet and exhaust, is included

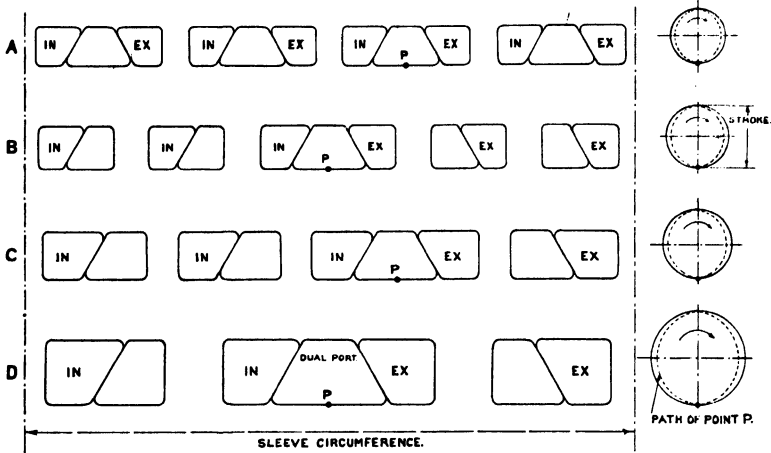



FIG. 72.
Arrangement and number of ports for semi-rotary sleeve valve.

A, four inlet and four exhaust ports in cylinder with four dual purpose ports in sleeve.
B, three inlet and three exhaust ports in cylinder with five ports in sleeve.
C, three inlet and two exhaust ports in cylinder with four ports in sleeve.
D, two inlet and two exhaust ports in cylinder with three ports in sleeve.

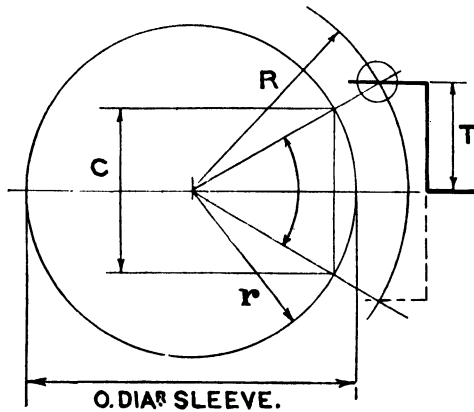



FIG. 73.
Geometry of crank movement in the operation of the Burt McCollum semi-rotary sleeve valve.

Chordal travel $C = \frac{2\,T \times r}{R}$

Where T = crank throw in inches.
r = outer radius of sleeve in inches.
R = distance from axis of sleeve to axis of pivot or ball-and-socket coupling in inches.

in each arrangement. This allows the largest area of ports for a given circumference of sleeve. The port area can be further increased by using more than one dual-purpose port, but the disadvantage is the necessity for alternate sequence of inlet and exhaust pipes round the cylinders. One dual port is therefore found to give the best compromise. The stroke of the crank for each arrangement is indicated by the major diameter of the ellipse shown to the right of each arrangement. The course taken by each spot on the sleeve in relation to the wall of the cylinder follows the elliptical path. The minor dimension of the ellipse is easily obtained by a simple geometrical layout as shown in Fig. 73. The loss of the full stroke from the crank is due to a translation from a linear to a radial movement, and the exact travel will depend on the distance between the spherical bearing or hinge pin and the periphery of the sleeve.

It is claimed that the elliptical path has advantages as it gives maximum sleeve speed at exhaust opening and inlet closing points, and minimum speed at inlet opening and exhaust closing. Although the actual variation in speed is small, the resulting characteristics are favourable to improved breathing.

CRANK THROW. The throw of the sleeve crank is obtained from the number of ports in the cylinder as given by the port arrangement selected (Fig. 72) and the dimensions D and R when:

D = outside diameter of sleeve in inches.
R = distance from axis of sleeve to axis of pivot pin or ball-and-socket coupling in inches.
T = throw of sleeve crank in inches.

Dimension R should be kept as small as possible and is generally 0·575 D when the ball-and-socket is used.

$$\text{Then } T = \frac{\pi D \times 0{\cdot}575\, D}{D\,(2\ \text{No. of ports}) - 1} = \frac{1{\cdot}8\, D}{(2\ \text{No. of ports}) - 1}$$

Example:

Port setting—three inlet and three exhaust in the cylinder.
Bore of cylinder—3 in.
Thickness of sleeve—0·125 in.
Outside diameter of sleeve D = 3·25 in.
Distance from axis of sleeve to axis of pivot R = 0·575 × 3·25.

$$\text{Then throw of crank} = \frac{1{\cdot}8 \times 3{\cdot}25}{(2 \times 6) - 1} = 0{\cdot}531 \text{ in.}$$

Much valuable information on the calculation of port areas and determination of valve settings, together with an analysis by Mr.

Burt, can be found in Ricardo's standard work on high-speed engines.*

Application to Aero Engines

Mr. Harry R. Ricardo is largely responsible for bringing into use for aircraft the semi-rotary sleeve valve engine. About 1922 research into the possibilities was undertaken with the support of the Air Ministry, who financed the work under a research contract. The career of the Burt invention up to that date, as is well known, had not been a happy one, but Ricardo considered that it had been shadowed by ill luck and commercial mismanagement rather than by any inherent technical faults. He was attracted to this valve for the attainment of very high outputs for three reasons:

(1) The head, being without valve pockets, left complete freedom in the design of the combustion chamber.
(2) The absence of a hot exhaust-valve head would reduce the tendency to detonate.
(3) The absence of valve springs removed at least one limitation on the speed of operation.

A large number of mechanical difficulties were encountered during development, but none proved formidable, and by 1924 a 50-hour Air Ministry test on a large single-cylinder unit, at an output and efficiency far beyond anything that had been attained at that date, had been successfully completed. The absence of an exhaust valve and the central position of the ignition enabled the compression to be raised by one whole ratio before detonation intervened.

On the strength of these results the design of a full-sized aero engine soon followed. Today the success of the semi-rotary sleeve is exemplified in three front-rank aero engines, the Bristol Hercules, the Napier Sabre 24-cylinder engine, and the Rotol auxiliary power plant. On each of these engines we find, in the design of the valve and its operating gear, the modern equivalent of the Burt principle. A description of the Rotol engine in Chapter VIII will illustrate by way of example the improvements that have been made since the demise of the old Argyll, Picard Pictet, and other pioneer manufacturers.

Important changes in materials and detail development of the operating mechanism have come about. Cast-iron pistons and cylinder barrels with sleeves of cast material were partly responsible for low heat dissipation, leading to local high temperatures, both

* *The High Speed Internal Combustion Engine*, by Harry R. Ricardo. Blackie & Sons, London, 1931.

grave hazards to successful lubrication. A partial or transient seizure of a sleeve was sufficient to overload the driving gear, resulting in mechanical failure of the components.

Freedom from seizure has been secured by machining the sleeve and cylinder to finer tolerances. This results in working clearances nearer to the ideal dimension required for satisfactory lubrication. The Bristol Aeroplane Company has given great attention to this and other matters and in 1943 patented a process for treating the faces most liable to seize up in some operating conditions. On the circumferential wall of the cylinder head, near the end which lies within the cylinder, is formed a helical groove having at least one complete turn. The depth of this groove is from 0·004 in. to 0·008 in., the width about 0·10 in. and the pitch about 0·025 in. or slightly more. The provision of the groove on the sleeve instead is equally effective in diminishing the risk of seizure.

A ductile bearing-metal having a low melting point can be used as a filling for the groove, applied in the following manner. Prior to completing the machining of the surface of the cylinder head or sleeve the grooves are formed upon it, a film of the bearing-metal is then electro-deposited all over the surface, so that it fills the groove and the part is then machined to its final dimensions. This removes the film of bearing-metal from the surface but leaves the indentations filled with it. Cadmium, which has a melting point of 320°C., is highly ductile and is a satisfactory metal to use. Lead, with a melting point of 327°C., or tin, with a melting point of 232°C., can also be used. Alloys of these metals having the required physical characteristics of ductility and melting point can also be employed.

The introduction of aluminium alloy for pistons, cylinders and heads has improved the heat transfer. Attention to other details and the incorporation of the hardened steel sleeve, less in thickness and less in weight than cast iron, are also factors responsible for enhanced reliability and the ability to maintain high m.e.p. for long hours without signs of sleeve seizure or distress.

CHAPTER VII

MODERN ROTARY-VALVE SYSTEMS

The Cross Rotary Valve

THE full development of the Cross rotary-valve system, with "controlled loading", is one of the outstanding technical advances of our time. The achievement is comparable with the invention of the Michell thrust block, although perhaps not of such universal application.

The commercial designs here described are the result of continued experimental work, tests and trials, since the early examples of Cross engines given in Chapter V.

In its present form the engine is easy to make and assemble; it is reported to be very reliable and free from any need of adjustment, as the valve is entirely automatic in taking up any slight wear. The power is smooth and the torque good throughout the whole range of speed. The heavy loading and indeterminate friction of the early models have been eliminated by the successful application of the novel principle of controlling the forces which apply load to the rotor, and the whole arrangement is amenable to rigid calculation.

The leading points in the design of this rotary valve which demand attention are (1) controlled valve loading, (2) circulatory scraper lubrication and (3) port edge sealing. These features are, of course, covered by patents. A pictorial view of the engine in section is given in Fig. 74, which indicates the principal components.

CROSS SYSTEM OF CONTROLLED VALVE LOADING. The actual loading of the valve is controlled by the designer so that a correct and sufficient pressure is exerted upon the valve to maintain its contact with the sealing port edge lip. As the pressure rises in the cylinder it endeavours to press the valve away from the port, but always the valve is given a little more pressure to seal the port than the gas can exert upon it to press it away.

The housing is divided horizontally and the two halves are always in resilient contact with the valve, which merely acts as a spacing member between them. If the rotor heats up more rapidly than the surroundings, and therefore expands more, the housing automatically adjusts itself to accommodate the rotor with no increase of pressure, and by virtue of this the expansion problem,

which has been a difficult one with all cylindrical rotors, is eliminated.

The valve is so lightly loaded at all times that seizure is most unlikely, and engines have been run dry for considerable periods without picking up. Needless to say, such a practice is to be

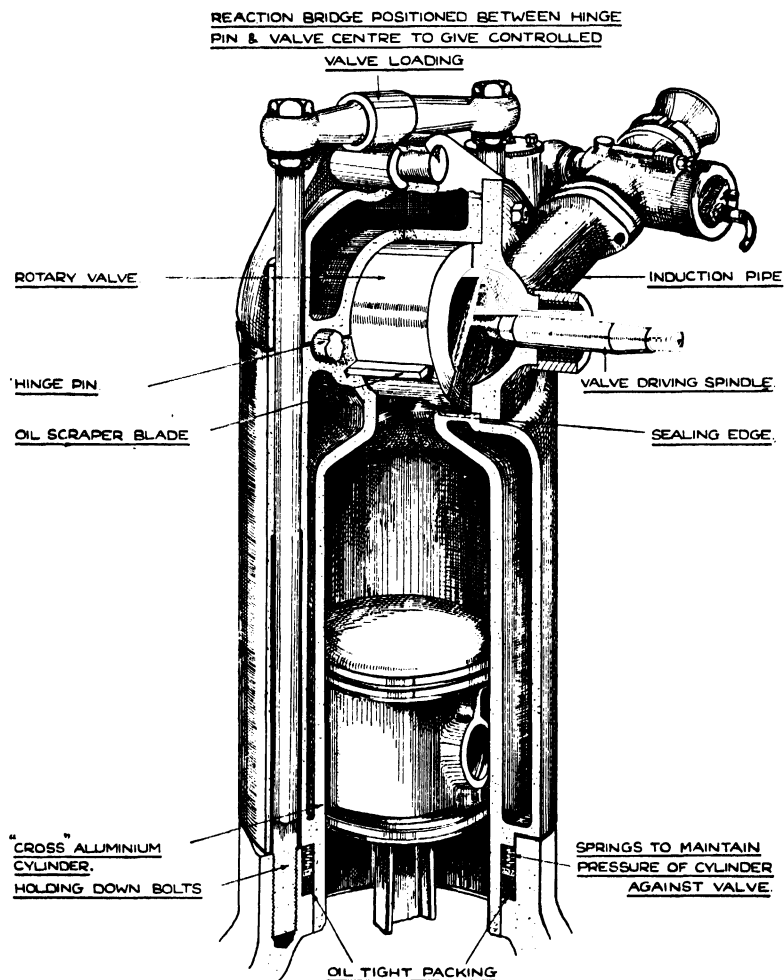



FIG. 74.
Pictorial view of Cross engine in section, showing the principal components.

condemned, as the valve and housing would wear very considerably under these conditions. In the normal running condition, however, when properly lubricated, wear is almost negligible, and what little wear does occur is automatically taken up, because the two halves of the housing maintain constant contact with the valve.

General practice is to use an aluminium cylinder head and to bore the hole in it to accommodate the valve. Thus, having removed the necessity for a valve bush, the designer is at liberty to place the valve scraper blade of the oiling system in the most convenient position, as there is no bush to sever in the process. By these means the mechanical efficiency of the valve is very high, due to the light loading and improved lubrication. Lighter loading in turn produces a cooler running valve, as less heat is generated by friction. Also, as there is no clearance between the valve and

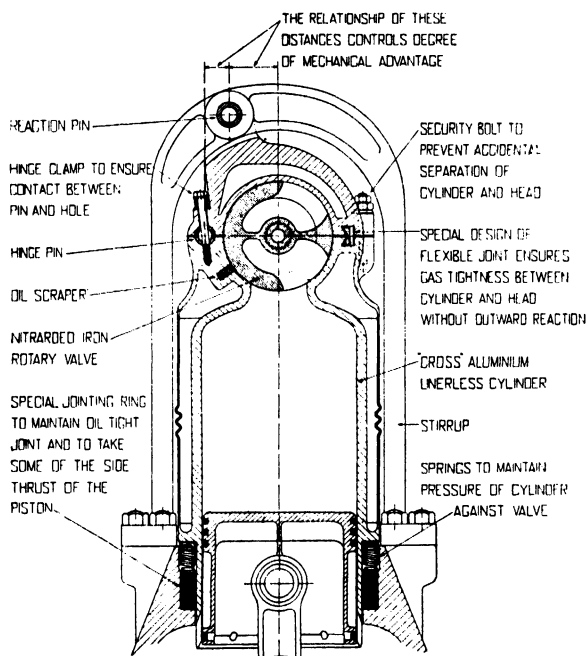



FIG. 75.
Cross rotary valve engine with controlled loading to ensure sealing of the valve without undue surface friction.

housing except for an oil film, the thermal contact is excellent and the valve can more readily dissipate its heat.

Controlled valve loading eliminates still another bad feature of the surface valve, namely the adverse influence of liquid petrol when starting from cold. In the Cross engine it is possible to pour petrol freely into the induction pipe of the engine and start up without any ill effects whatever. Again, it is not necessary to warm the engine up carefully before calling for full power. The engine can be driven away at speed from cold with no disadvantages whatever because the thermal expansion problem no longer exists.

Still a further advantage is to be found when starting from cold. The self-adjusting contact system of the controlled valve loading ensures that the rotor is free to turn under all conditions and an easy start can be accomplished in the coldest weather.

To describe the system in greater detail it is necessary to refer to Fig. 75, from which it will be clear how the modified controlled loading is brought about. This is effected by utilizing the gas pressure in the cylinder to provide the necessary force to exert pressure on the two halves of the valve housing, and limiting the amount of this force by the principle of leverage or mechanical advantage. The cylinder unit is allowed to float axially, and to prevent it leaving the crank chamber it reacts against an overhead stirrup gear. The rotor is trapped between the bottom half of the housing, which forms part of the cylinder block, and the top half of the housing. As the pressure in the cylinder rises, the greater will be the force reacting on the stirrup gear, and if uncontrolled this force would exert far too heavy a bearing-load on the rotary valve, particularly in large engines where the valve is small in relation to the diameter of the bore.

The mechanical advantage or leverage to reduce this pressure can be brought about in a variety of ways. A pair of hinged levers can be fulcrumed on the stirrup, one end of each lever reacting on the upper half of the valve housing, the other against the lower half. A simpler method, and one which works well in practice, is to use the top half of the valve housing as the lever and insert a hinge pin at one of the joints between the two halves of the valve housing. The upper half of the valve housing is attached to the overhead stirrup at a position between the centre of the valve and the hinge pin. The ratio of the distances between the hinge pin and the reaction point, and the valve centre and the reaction point, is the controlling factor determining the proportion of load which is applied to the rotor and the proportion which passes directly through the hinge pin to the stirrup.

In some engines it has been ararnged that this reaction point can be varied while the engine is running, and it is most interesting to observe how the torque rises appreciably as the reaction point approaches the optimum position. It is, of course, easily possible to calculate the correct position, and in doing so the designer must take account of the ratio of the port area to the bore area, as it is obvious that a proportion of the explosive force, depending on the area of the port aperture, is passed directly to the top cap by way of the rotor. It is important to note that this proportion is a direct load on the top cap, and does not come under the influence of the controlling leverage except to act in a contrary direction, and its effect must therefore be included in the calculations when the designer is working out the required sealing pressure on the lip.

It will be clear from the above that the total load on the top surface of the rotor is of greater magnitude than that on the sealing side. Since the valve/piston area ratio is usually about 0·21, the load on the top surface is approximately one and a quarter times that on the bottom surface.

This condition is somewhat of a disadvantage, but it is inherent in the design. In fact, it must be fully understood that more than 50 per cent of the friction load is not utilized for sealing purposes at all, and is therefore a prime tax on the mechanical efficiency and a source of power-loss.

It has been found that an engine will work when the sealing lip is pressed against the rotor by a force of only 1 per cent of the explosion pressure in excess of that necessary to ensure an effective seal, but it is usual to design for sufficient force to seal plus a margin of 5 to 10 per cent for a safety factor, and also to make quite sure that there is no possibility of the two halves of the valve housing being separated by gas pressure. Precautions are also taken in the design to prevent accidental opening of the two halves of the valve housing, and adjustable stops are provided to ensure that such separation could only be of small magnitude.

It is necessary, of course, to provide some light spring pressure underneath the cylinder to maintain the upward pressure of the cylinder when there is no gas pressure available in the cylinder, as, for instance, during the induction stroke. This, however, adds but little to the friction, because the load of the springs is controlled by, and in the same way as, the cylinder pressures.

Cross Circulatory Scraper Lubrication. The satisfactory lubrication of the rotor is of utmost importance, as the mechanical efficiency of the valve gear depends so largely upon it. The film of lubricant also plays an important part in the passage of heat from the rotor to the valve housing, and during the course of its travel a small amount of heat is taken up by the oil and dissipated during its return journey to the source of supply.

The system will be understood by reference to Fig. 76. The oil is pressure-fed on to one side of the valve and is then carried round by the rotor through about 220 degrees of travel, when it meets the scraper blade, which is resiliently mounted in a slot in the valve housing so that it presses with a sharp edge against the surface of the valve. The oil then leaves the valve face and enters the scraper duct, except for the film which adheres to the face of the valve. This is infinitesimally thin, but nevertheless sufficient to lubricate it during the remaining 140 degrees of travel until it reaches the oil inlet groove. Situated at the back of the scraper blade is a non-return valve, which is a very important feature. The system will not work satisfactorily without this component. At each revolution of the valve, first the exhaust and then the inlet port

pass over the scraper blade, and it is by using the pressure-difference in these two channels that the oil is forced back to the source of supply. When the exhaust passes the scraper blade the exhaust pressure opens the non-return valve and a very small quantity of gas passes down to the oil discharge pipe and in so doing carries with it the oil which has accumulated upon the scraper blade in one revolution. The inlet port now registers with the scraper, and due to low pressure in this channel the oil would be sucked back

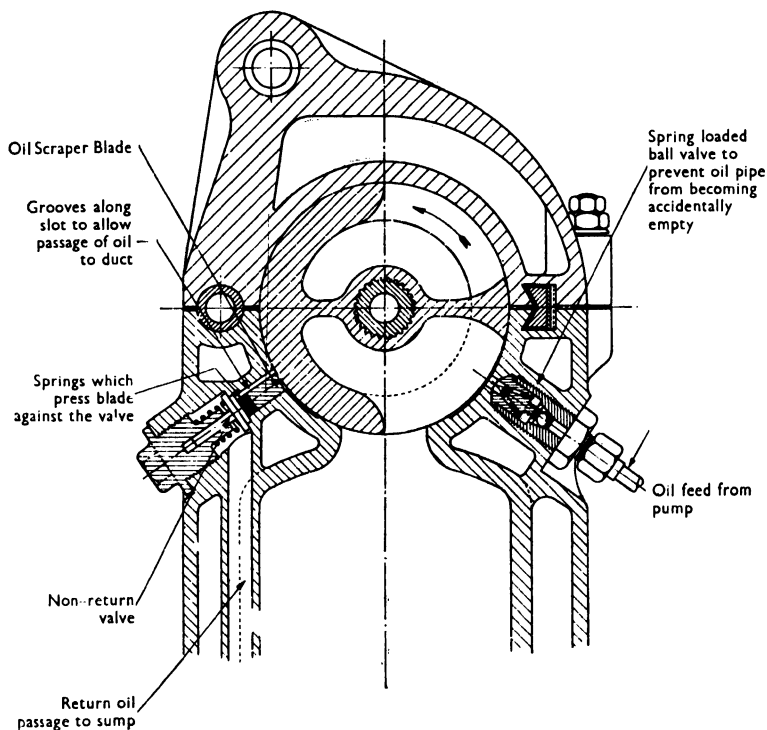



FIG. 76.
Lubrication system of Cross rotary valve with modern construction of the scraper and non-return valve.

again but for the fact that the nor return valve is drawn on to its seat and prevents this occurring. It has bee[illegible] in practice that a second no[illegible]eturn valve near the [illegible]arge pipe adds to the efficiency of the [illegible]

The scrap[illegible] bla[illegible] and [illegible]ve parts are very simple to make and the m[illegible] cylinder block to accommodate them prese[illegible]ts no undue difficu[illegible]y. The best results are obtained by using a positive action plunger pump and controlling the supply of oil to the valve by a linked [illegible]onnection with the throttle control.

By this system the oil is fed to the valve in proportion to heat and load conditions, and a greater economy in oil results.

The scraper blade and the non-return valve, being constantly supplied with clean oil, remain in good condition and do not carbonize or gum up. The exhaust is smokeless and oil dilution tests have been made and found to be the same as in the average conventional engine.

RAISED LIP PORT-EDGE SEALING. The poppet-valve has generally possessed some advantages in gas tightness over the rotary valve. The Cross rotary valve is manufactured to fine limits and is gas tight from the time the engine is first assembled, and it does

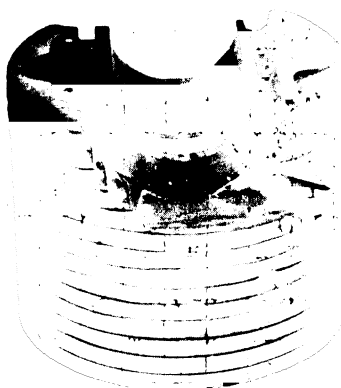



FIG. 77.
Air-cooled engine cylinder with Cross raised sealing lip around the port edge.

not require carbon deposits to close up sources of leakage. Gas tightness remains equally good for the whole life of the engine, from the fact that wear is automatically taken up as explained previously.

The actual seal takes place at the mouth of the port due to the raised sealing lip, as shown in Fig. 77. The lip is formed by removing the [illegible] for the length of the valve and up to about ten degree[illegible] in the valve housing. The depth of this recess is only [illegible]

Both the surface of the va[illegible] [illegible] d sealing edge work at a relatively low temperature [illegible] do not get burned. Both surfaces remain bright, smooth and polished and there is always a thin film of lubrication between them. As the inlet gases pass through the same cylinder port hole as the exhaust gases, a certain

amount of cooling of the port edge takes place which tends to maintain the good condition of the lip.

Experience shows that the sealing property of the Cross rotary valve is superior to a normal piston ring seal. In support of this statement a special cylinder unit was made to test the gas-tightness of the seal on the rotor. In the cylinder was fitted a piston having a large number of compression rings, the idea being to create varying air pressure by moving the piston up the bore. It was

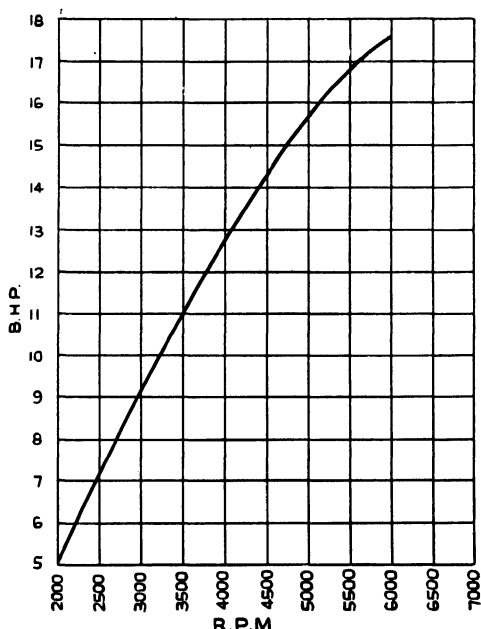



FIG. 78.
Power curve of Cross 247 *c.c. air-cooled engine up to speed of* 6000 *r.p.m. with exhaust fully silenced. Cooling wind speed* 71 *m.p.h. Height of test house* 550 *ft. above sea level. Barometer* 29·6 *in. of mercury. Fuel* 65 *octane—calorific value* 139,200 *B.T.U. per gal.*

found, however, that the compression in the cylinder could be held for only a relatively short time and there were no indications of escape past the valve. It was then found that all the air was escaping past the piston, and this extraneous leakage could only be cured by operating the cylinder in an upright position with a layer of thick oil over the top of the piston crown.

The sealing lip, it is stated, is not susceptible to wear, and will last almost indefinitely when run against a nitrided valve. The lip is in most cases formed in the same metal from which the head

is made and the latter is normally an aluminium casting such as Y alloy.

PERFORMANCE CHARACTERISTICS OF CROSS ENGINES. The Cross engines combine a remarkable degree of power with the smoothness and flexibility of engines of much lower efficiency. Brake mean-effective pressures of 165 lb. per sq. in. are easily obtained with normal aspiration on a fuel of only 65 octane. Judged by normal

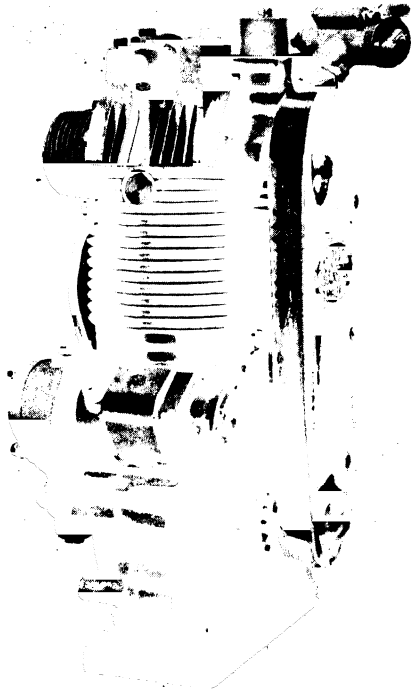



FIG. 79.
Cross typical motor-cycle engine. Bore 70 *mm., stroke* 90·5 *mm., cubic capacity* 348 *c.c., compression ratio* 10·6 *to* 1.

standards, it would be expected that such an engine would be exceedingly rough, subject to detonation and somewhat inflexible, but this is not so and the engine will work hard at low speed with a good low-end torque. Using fuel of 80 octane, a b.m.e.p. of 195 lb. per sq. in. has been attained at 4000 r.p.m.

The high performance and low fuel consumption are considered to be the result of the cool combustion chamber which allows high compression ratios to be used. Great care has been taken in the

Cross design to ensure that no hot spots can occur and that ample heat flow is provided from all parts of the combustion chamber. The rotor, being lightly loaded, does not gain much heat from friction, and as it is in constant and complete contact with the valve housing, heat dissipation is good. The influence of this last feature upon performance is considered important, and it has been found by experiment that the hotter the valve the lower must be the compression ratio, with a corresponding lower performance.

A typical horse-power curve is shown in Fig. 78. This represents the figures obtained on test of a 247 c.c. air-cooled motor-cycle engine of the type shown in Fig. 79, with a compression ratio of 11 to 1, and it will be seen that the horse-power does not fall off appreciably at a speed of 6000 r.p.m. The petrol consumption at an observed speed of 4000 r.p.m. equals 0·45 pint per b.h.p./hr. This figure has been improved on later engines down to 0·41 pint per b.h.p./hr.

It is claimed that periods between necessary decarbonization are much longer than with the corresponding poppet-valve engine, and the performance does not fall away with the formation of carbon, as is not unusual in the conventional poppet-valve engine.

Hinge Pin Construction. Several methods of construction have been utilized in carrying out the system of controlled loading, and in Fig. 80 is seen one of these variants as applied to a motor-cycle engine, wherein the axis of the rotor is placed adjacent to the combustion chamber, but to one side of the vertical centre line of the engine. The cylinder in this arrangement is positioned and held relative to the crankcase by a horizontal hinge pin passing through a boss formed integrally with the cylinder and approximately one-quarter of the way up the stroke. This pivot construction replaces one of the two tie bolts which held down the cylinder in the basic design first described, and it will be seen that this pivot locates the cylinder with great accuracy in every direction except one and restrains the line of freedom to a radial path about the hinge pivot. The pressure within the cylinder, therefore, tends to move the cylinder in a clockwise direction, and the reaction is taken, through the rotor housing, on the abutment bracket shown to the right of the valve. This bracket is bolted rigidly to the lower part of the crankcase.

The loading on the valve is therefore reduced by the ratio of the vertical arm to the horizontal arm, in this case approximately two and a quarter to one. In other words, the proportion of the piston force which is applied to the rotor is governed by the moments around the hinge pin.

A spring or a plurality of springs is interposed between the base of the cylinder and the top of the crankcase on the side of the barrel remote from the pivot. By these means the seal is at all

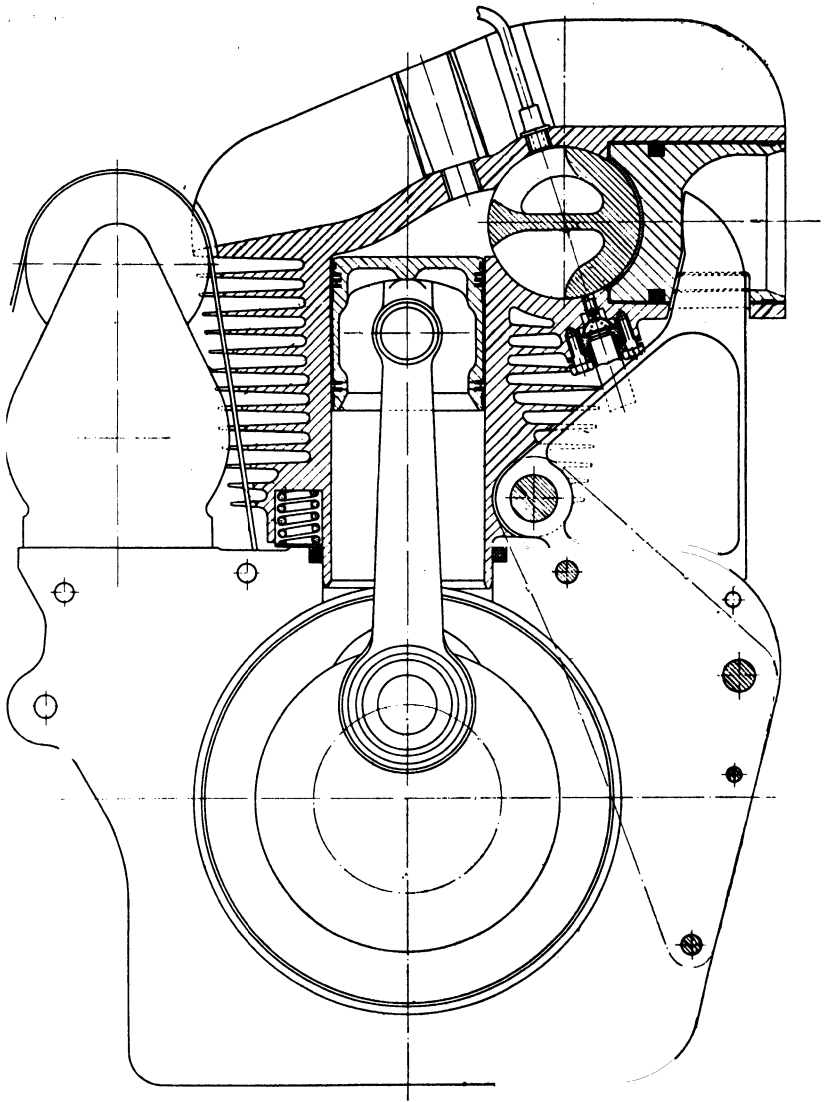



Fig. 80.
Cross engine with hinge-pin cylinder mounting and rotary valve at side.

times in close contact with the rotor, and any slight wear of the working surfaces is automatically taken up, in the same way and with the same effect as in the basic arrangement, where the rotor is disposed centrally above the combustion head.

Cross Light Aero Engine. It is of general significance that although a considerable number of successful single-cylinder rotary-valve engines have been built up to the present time by several individual constructors, there has been less evidence of successful application to multi-cylinder engines. It is therefore of some interest to describe the 150 h.p. light aero engine designed by the Cross Manufacturing Co., Ltd., of Bath. This is a four-cylinder unit of 100 mm. bore by 100 mm. stroke, with a cubic capacity of 3141 c.c. and a normal speed of 4500 r.p.m.

The loading of the valve in this example is governed by a compound system of leverage applied in an ingenious manner, but without departing from the basic principle of control. The mode of effecting the control will be understood by reference to Fig. 81. Firstly, taking moments round the hinge pin on the left-hand side of the cylinder, the load in the vertical tie bolt to the right of the cylinder centre line is found. Let R = reaction in the tie bolt and W = load on piston. Then in the present instance:

$$R = W \times 3{\cdot}30/4{\cdot}75.$$

This represents the downward force on the keep, a proportion of which is transmitted to the rotor and the balance to the spherical button to the right of the head which acts as a fulcrum. Secondly, to compute the share of the load borne by the rotor, R must be multiplied by the horizontal distance between the bolt and the fulcrum, and divided by the distance from the rotor axis to the same fulcrum. Then it follows:

$$R \times 1{\cdot}87/3{\cdot}375 = \text{load on top side of valve.}$$

Thus the magnitude of the load borne by the rotor is considerably reduced in relation to the full load on the piston.

The load on the sealing side of the rotor may be calculated in the same manner, except that the area of the port orifice must be subtracted from the area of the piston to arrive at the nett upward load W, before applying the effective leverages for the purpose of evaluating R. If, as is approximately the case, the port area is 0·20 of the piston area, then the relative values of the upper and lower loadings will be as five is to four approximately.

In the design of a multi-cylinder engine there are several problems to be solved which do not occur in a single-cylinder unit, and the Cross arrangement is no exception to the rule. The hinge pin construction allows freedom to each of the cylinders in an upward radial direction about the axis of the hinge pin, and since the movement is limited by the adjustable tie bolts, the cylinder heads with their keeps can all be lined up to the valve

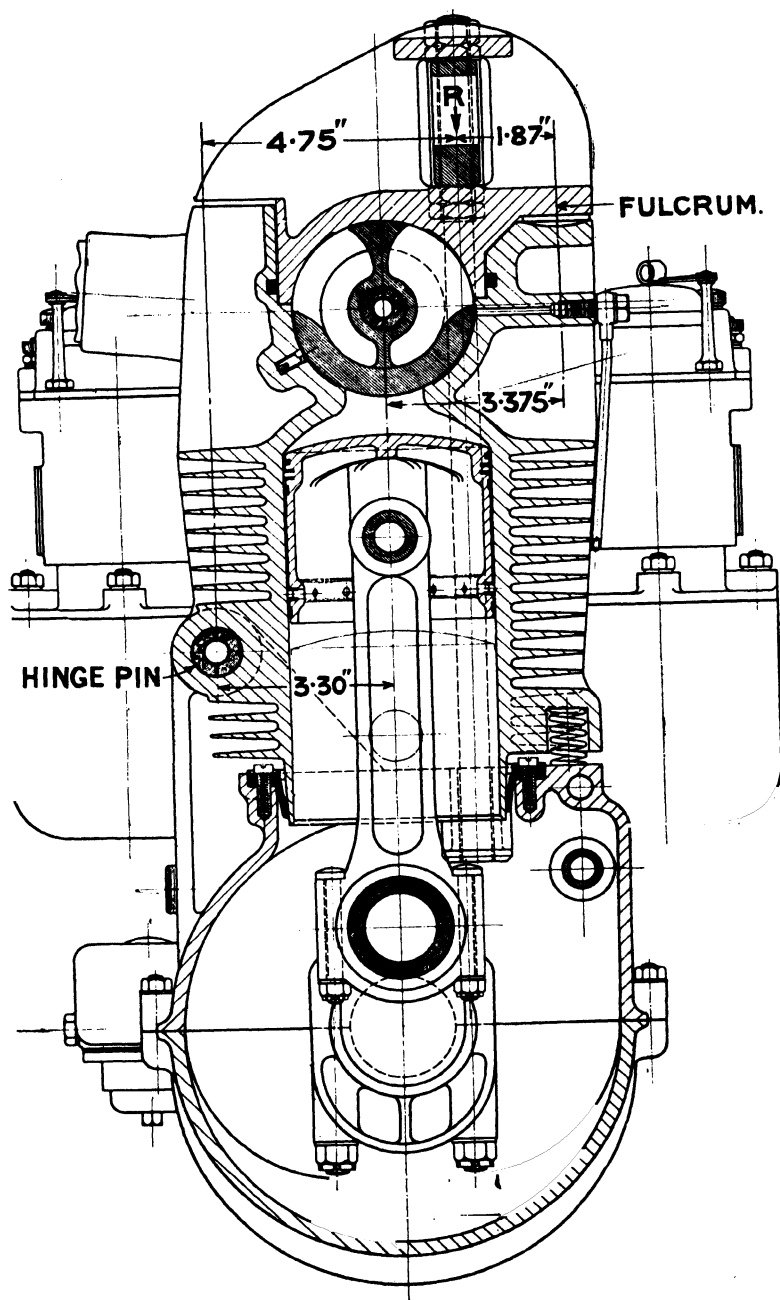



FIG. 81.
Cross 150 h.p. light aero engine with compound system to reduce the valve loading.

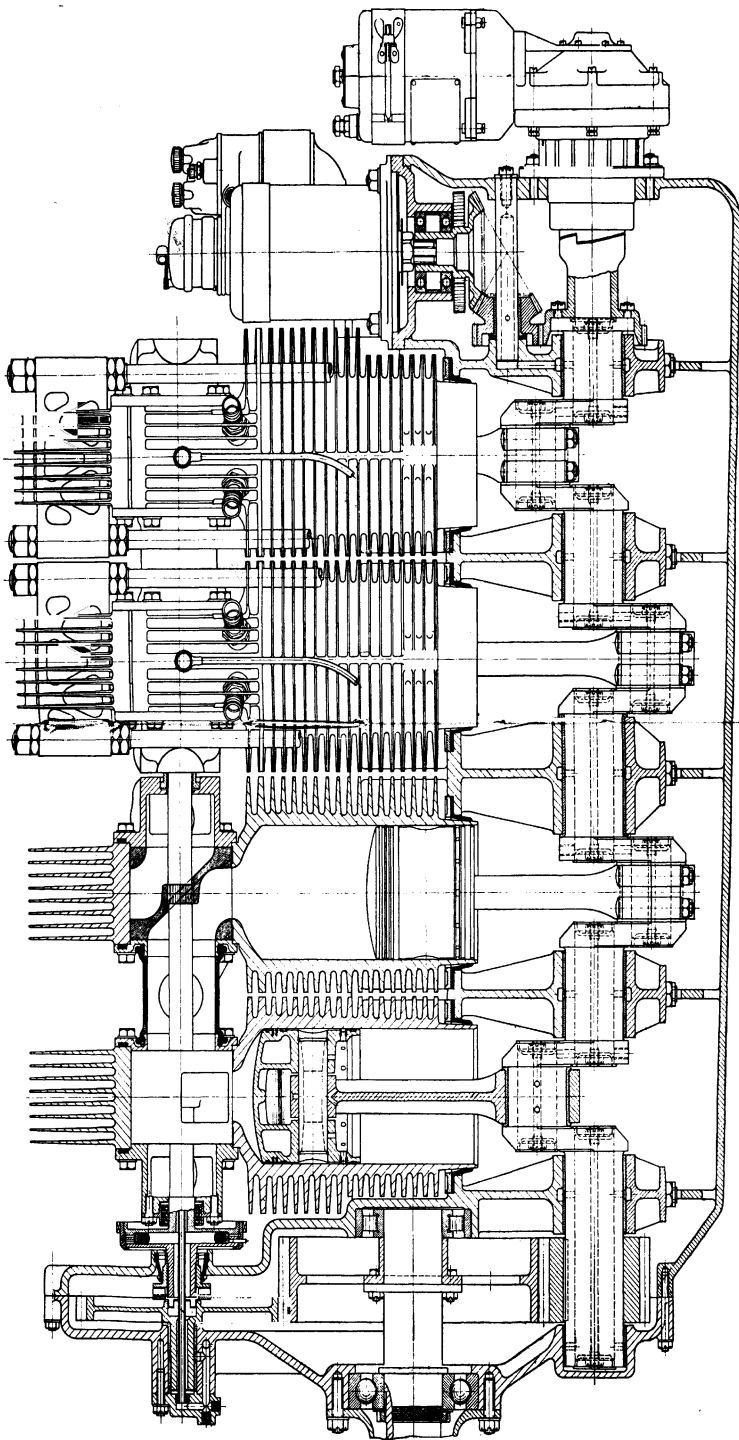



FIG. 82.
Longitudinal elevation of Cross 150 h.p. light aero engine.

shaft which drives each one of the several rotors through the medium of a splined fit, as shown in Fig. 82, which is a part longitudinal section through the engine. This is obviously an excellent system so far as the valves are concerned, although requiring a high degree

of accuracy in the adjustment of the tie bolts. It introduces, however, another problem. The manifold connections between cylinders must necessarily be of a flexible nature, so as to allow the very slight freedom needed in order that the sealing component may

function. This requirement is met by fitting a short resilient pipe between adjacent cylinders. The construction indicated in the drawings has been found to be effective, and it is of universal

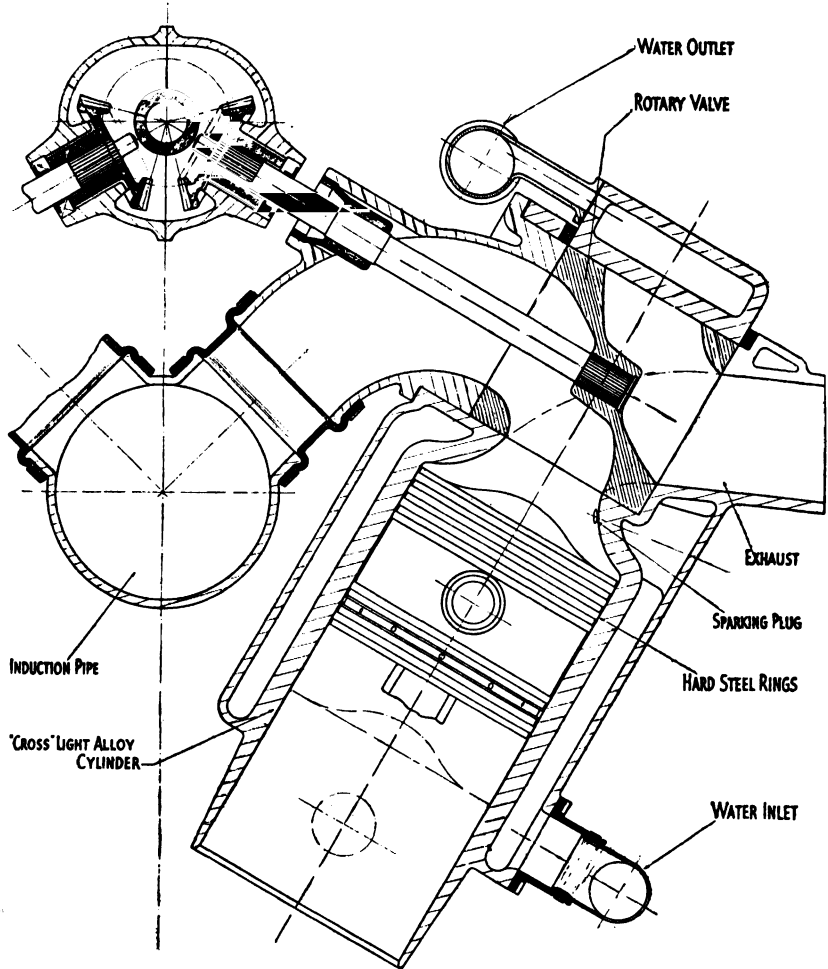



FIG. 83.
Application of Cross rotary valve to V engine of the "Merlin" type.

application to multi-cylinder engines with banks of cylinders in line and in other formations.

APPLICATION TO V ENGINES. As an example, Fig. 83 shows how the system is applied to a V engine of the Merlin type. Again, each cylinder is an independent unit and the branch inlet pipes are joined to the induction manifold by a similar type of flexible

connection. The cylinders in this layout of the Merlin equivalent are provided with water jackets. The keep over each rotor is also adapted for liquid cooling.

As is generally the rule in arrangements of multi-cylinder engines with a separate rotor to each cylinder, the drive becomes somewhat elaborate, and it should be noticed that in this design a certain amount of flexibility is also required between the drive-gears and the rotor. This has been carefully provided by the introduction of a resilient bearing at the point where the splined shaft leaves the intake branch, adjacent to the bevel gear box. These complications have their disadvantages but cannot be avoided where perfect freedom to the rotor is a requirement of the system.

In none of these engines has water cooling been applied to the inside of the rotor, since in small and moderate sized valves it becomes an unnecessary complication. The Cross Manufacturing Co., however, have shown in some of their early engines how

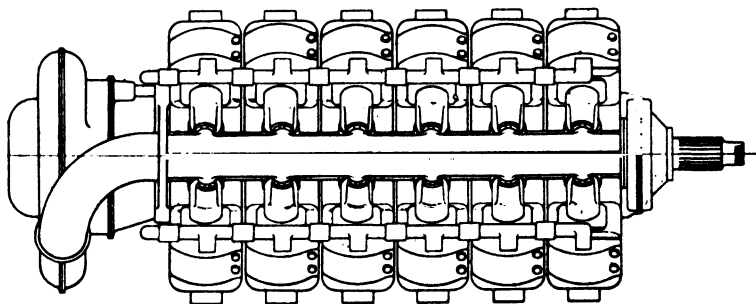



FIG. 83A.
Plan view of the V engine.

liquid cooling is carried out and have stated that if at any time they were called upon to design a large Diesel or spark-ignition engine, they would most likely provide for water circulation through the rotors.

Aspin I.C. Engines with Rotary Combustion Chamber

A great amount of mechanical and metallurgical work has gone into the development of the Aspin production designs since the first published descriptions of the prototype engines in 1937. The thermal efficiency of the petrol engine now challenges the consumption figures of the oil engine, and on a gallonage basis it is able to compete with the compression-ignition oil engine and at the same time provide a smaller, smoother unit with a wider range

of speed and less overall weight, thus conducing to all-round economy.

The Aspin rotary valve or, using the term preferred by the producers, the "rotary combustion-chambered" engine, gets nearer to theoretical ideals than any prime mover with conventional poppet-valves. It is claimed that the rotary unit requires less skilled attention than is demanded by the average exhaust poppet-valve, and that it has more positive valvular action, higher efflux values, and greatly reduced local heat, thus improving the knock-resistance quality and raising the upper limit of mechanical operation. Some of these qualities have also been attributed to sleeve valve engines, but the rotary combustion chamber is claimed to have functions other than purely valvular.

At the time the first experiments were made, the features of knock resistance, high speed of rotation and fuel economy encouraged further development. Various existing records of power, speed and efficiency, both thermal and overall, of unsupercharged engines were surpassed.

Functions of the Rotary Combustion Chamber

Since then work has been devoted to harnessing and adapting to more useful purpose the principles which the prototype embodied. The detailed shape of the head is of great importance, for upon it depend the functional characteristics. In a modern assembly the piston crown and the lower face of the rotor are machined to match, so that at T.D.C. only a very small working clearance exists between these two components. In consequence, almost the whole of the inspired charge is projected into the cell of the rotor, constituting the combustion space. This cell is machined by form cutter or other means to ensure that the shape and volume are closely defined and exactly duplicated for multi-cylinder engines.

Fig. 84 shows diagrammatically the operating cycle of one cylinder in four positions:

A. Induction—Cell registers with inlet port. Piston descending.
B. Compression—Cell sealed. Piston descending.
C. Firing—Ignition has occurred. Plug nearly screened. Cell sealed. Piston descending.
D. Exhaust—Cell registers with exhaust port. Piston ascending.

From these diagrams it will be noticed that the ports in the head are both closed by the solid portion of the rotor cone during

the compression and firing strokes, inflammation taking place in that portion of the cycle between B and C, Fig. 84, in a position remote from the hot exhaust port. It is claimed that this head goes further than other non-poppet-valve systems in reducing the temperature of parts or areas liable to overheating to which the live charge is exposed, by conveying the mass of the combustible mixture to the other and cooler side of the cylinder for combustion.

The knock-resisting feature is increased by other influences. It is agreed by some authorities, on the phenomena of detonation, that the more immediate and determining knock activator is the pressure wave preceding a defined flame front moving towards the area prone to detonate. Any means therefore to break up this front will assist in reducing detonation. The Aspin head effects

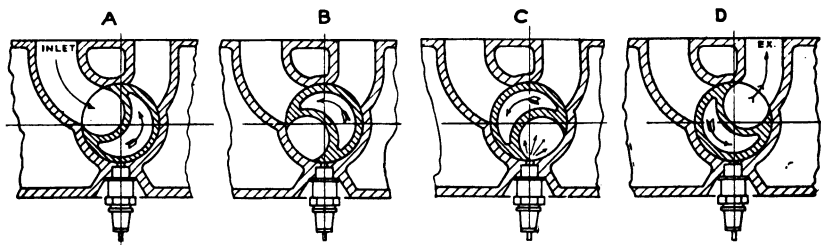



FIG. 84.
The position of the Aspin rotary valve at four stroke-positions of the engine.
A, Induction: Cell registers with inlet port. Piston descending. B, Compression: Cell sealed. Piston ascending. C, Working: Ignition has occurred. Plug is now screened. Cell sealed. Piston descending. D, Exhaust: Cell registers with exhaust port. Piston ascending.

this, as shown diagrammatically in Fig. 85, which represents an inside view of the combustion space immediately after the firing has been initiated by the sparking plug. The flame spreading from the spark meets the charge being projected towards it from an infinite number of radial directions as the piston squeezes the gases in the narrowing clearance between the piston top and the underside of the head. The hypothesis is put forward that for the reason stated a frontal definition of the flame is improbable, and the formation of a defined pressure wave unlikely to mature.

Another influence adverse to detonation, peculiar to the rotating head, is a centrifugally induced charge stratification. By centrifugal action the heavier constituents of the fuel charge will be thrown to the orbital periphery of the cell path; consequently, a relatively inert layer will remain near the inner wall of the cell, opposite to and remote from the plug.

The flame front moves towards the inner wall, where, ordinarily, terminal detonation would occur. In this case it is suggested that a quenching action takes place instead.

The sponsors therefore claim there are three knock inhibitors peculiar to the Aspin engine:

(1) Conveyance of the charge to the coolest part of the cylinder for combustion.
(2) Breaking up of the advance pressure wave.
(3) Providing an inert quenching layer for the terminal flame.

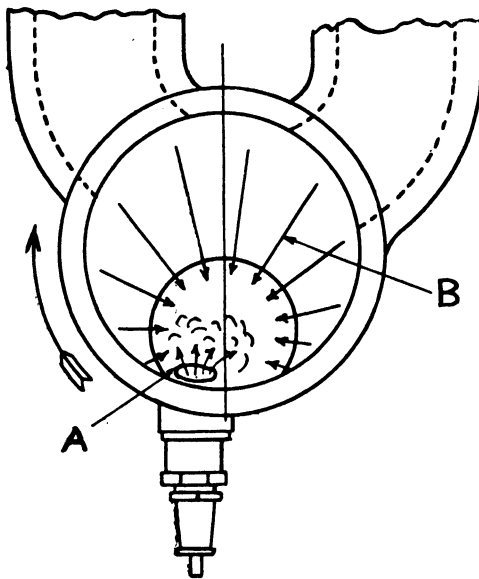



FIG. 85.
Diagrammatic inverted plan view of Aspin combustion space immediately after firing.

The respective importance of these three factors is a matter for speculation, but collectively their value is confirmed in the ability of the engine to work with low octane fuels at compression ratios of between 13 and 14 to 1.

The centrifugal stratification also appears to render possible the combustion of much leaner mixture ratios than is generally the case with spark ignition. Few standard type poppet-valve engines will burn economically a leaner air/petrol ratio than 16 to 1, but the Aspin head, at high engine speeds, will burn, and burn efficiently, mixtures as lean as 22 to 1. A probable benefit accruing from

centrifugal action is a progressively better scavenging by means of a positive tangential "fling" to the exhaust gases as the speed of rotation rises. A glance back at Fig. 84 D will show how this takes place. Although it is difficult to determine quantitatively the benefits derived from this phenomenon, there are indications that it has a favourable influence, making for improved scavenging.

Modern Air-Cooled Engine

A number of experimental engines have been made for the purpose of proving the theories expounded above. A few of the more recent milestones in design and development are illustrated. Fig. 86 indicates a modern arrangement in which the thrust bearing

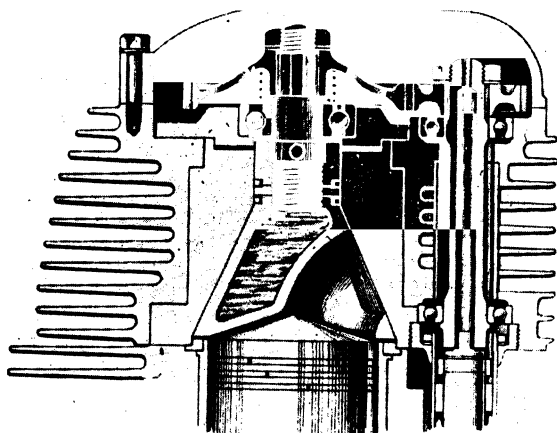



FIG. 86.
Modern Aspin air-cooled engine with entire load taken on the rotor surface.

has been discarded. The materials are representative of a present-day metallurgical combination. The rotor is of cast iron, oil filled to reduce weight; a special lead-bronze insert is assembled as a force-fit into the cast aluminium head. The entire load is taken on the conical surface of the rotor. The construction is reputed to give a life considerably longer than the remainder of the engine, and the performance is equal to a b.m.e.p. of 180 lb. per sq. in.

8 *h.p. Water-Cooled Engine*

A present-day design of water-cooled engine with four cylinders of nominal 8 h.p. is shown in Fig. 87. The four rotors are driven by skew gears from a horizontal shaft running the length of the engine. Each skew wheel on the top of a rotor is supported on two ball bearings, thus isolating any gear thrust reactions from the working surface of the rotor. The cylinder head is an iron casting,

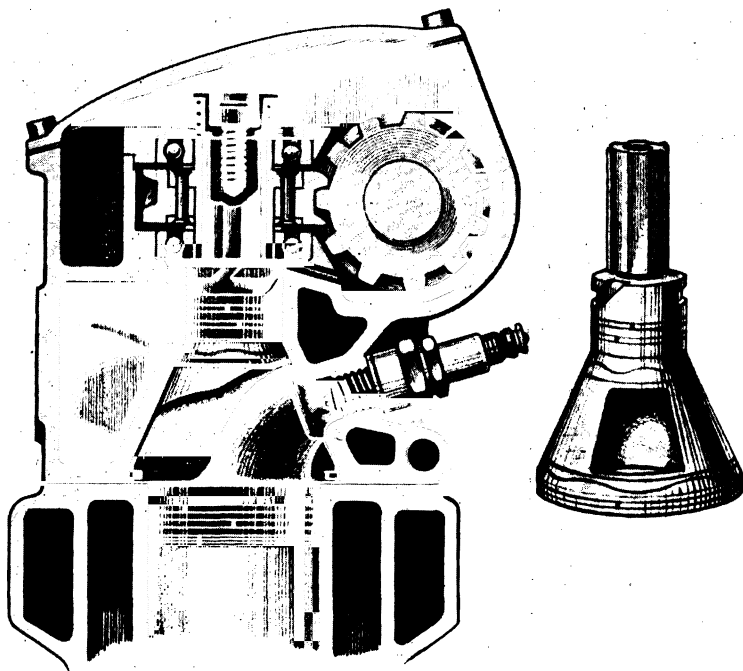



FIG. 87.
Present-day 8 h.p. four-cylinder water-cooled Aspin engine.

whilst the rotor is constructed from mild steel but faced with a coating of lead-bronze to provide a suitable mating combination for running direct on the counterpart cast-iron conical surface machined directly in the head.

It will be seen that the neck of the rotor is provided with a channel running partly round the circumference, for the purpose of affording an oil pressure timing device, which allows oil to pass to the rotor during only part of a complete cycle, and this avoids over-oiling during the suction stroke when there is a depression in the inlet pipe and cell. This feature improves oil consumption and prevents all possibility of smoke or fouling of plugs.

Heavy-Duty Aspin Engine of 4·6 Litre Capacity

One of the most recent fully developed commercial designs is shown in the general assembly, Fig. 88. This engine is suitable

[By courtesy of "The Automobile Engineer".

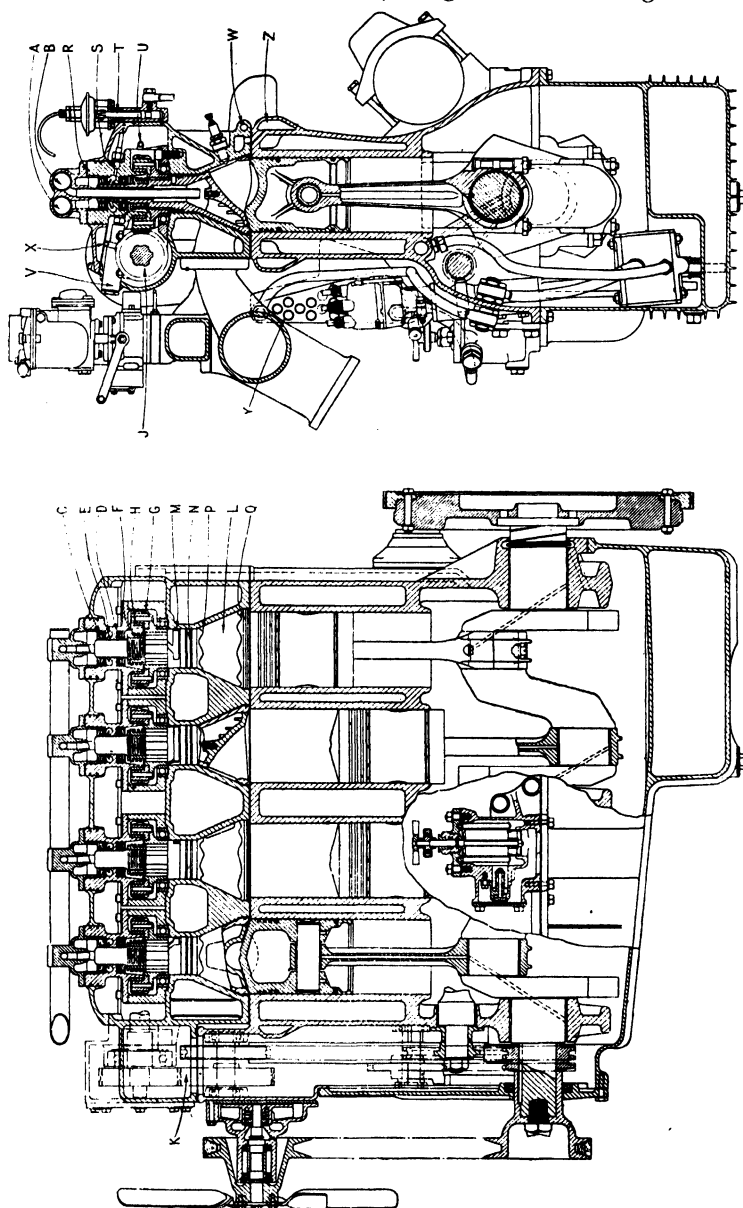



FIG. 88. *General assembly of Aspin heavy duty engine of 4·6 litre capacity.*

for heavy road traction. There are four cylinders of 4¼″ bore by 5″ stroke with a compression ratio of 9·2 to 1 and a design speed of 2500 r.p.m. As the engine develops a b.m.e.p. around 140 lb. per sq. in. as a maximum, the crankshaft is exceptionally robust. Made of 3½ per cent nickel steel, oil hardened and chromium plated on all journals, the weight of the finished shaft is 110 lb. The journals and crankpins are 3″ and 2¾″ diameter respectively and the connecting rods, of 65 ton carbon manganese molybdenum steel, are adequate in section to withstand the stresses imposed by the peak pressures resulting from the high compression ratio.

All other components and the general structure are of adequate strength in view of the relatively high loading, but it must be remembered that in spite of the high mean effective-pressures, the peak loads are not comparable with those resulting in the Diesel cycles, and in consequence the weight of the engine is considerably less. The cylinder head and rotary valve differ greatly in detail from previous layouts, although the basic principle of design remains the same. For commercial engine practice it was considered the uncooled solid rotor should be superseded by an internally liquid-cooled unit, and this is fabricated in steel as a simple means of obtaining a hollow component. It is faced with a special lead-bronze alloy and runs directly in a complementary seating in the cast-iron cylinder head, the conical surface carrying the thrust load. The roller thrust bearing is dispensed with, thus simplifying construction but demanding more from the metallurgical properties of the mating surfaces. The liquid cooling of the rotor lowers its general temperature, makes for smoother running and, it is claimed, lowers the flame rate.

Rotor Drive. The rotor is driven by skew gears from a horizontal shaft running along the top of the heads and parallel with the crankshaft axis. The drive is in two stages, by duplex chain from the crankshaft to a small countershaft below the level of the joint face and thence by a pair of spur gears to the overhead shaft to permit the easy separation of the head from the cylinder block.

Water Cooling. Fig. 89 shows the assembled cylinder head and rotor in vertical and horizontal sectional views, with the path of the coolant indicated by small arrows. Integrally cast projections amplify the contact area of the rotor closure plate exposed to the cooling water. The pump draws water direct from the radiator and delivers it to water galleries cast in the cylinder block. From these galleries jets of water impinge on the cylinder heads immediately below the sparking plugs and under the exhaust ports. Cylinder barrels are cooled by thermo-siphon from the cylinder head. The supply for the rotors is tapped off the main discharge from the

pump and rejoins the main system at the return from the engine radiator.

LUBRICATION. A feature of interest is that the lubrication of the rotor is controlled by a special valve pneumatically operated by the depression in the manifold, so that the oil supply is automatically increased with the load and vice versa. The oil regulating device and oil filter is in one unit. A piston valve, balanced by crankcase pressure at each end, has an annulus turned about its mid-section to furnish a means of communication between an oil supply port and a feed port on opposite sides of the bore in which it slides. The depression in the engine manifold, inversely proportional to

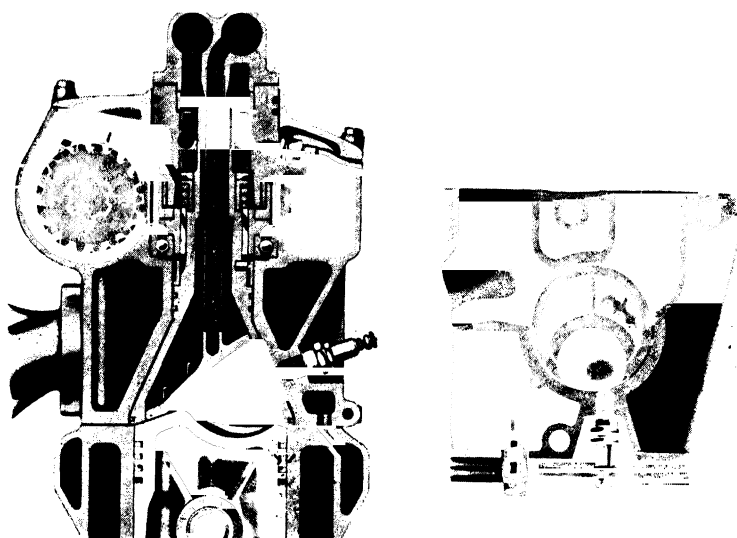



FIG. 89.
Aspin 4·6 litre cylinder head and rotor, showing the path taken by the coolant.

the m.e.p., influences a pressure-sensitive device coupled to the piston valve and so regulates the supply.

As the depression increases, the piston is unbalanced and moves to restrict the supply of oil to the gallery from which the rotors are fed. When idling with a closed throttle the supply is completely cut off. Immediately the throttle is opened the piston moves to open up the supply of oil, and this reaches the rotor bearing surface by way of a control groove in the rotor neck, which provides a timed feed as in the 8 h.p. engine already described. Fig. 93 indicates very clearly the path of the lubricant after it leaves the oil filter and regulating device.

CARBURATION. In poppet-valve practice the combination of valve sizes, lifts, timings, etc., with sparking plug types and carburettor settings for one set of requirements, is invariably unsuitable to other and different needs. The carburettor adjustment for a maximum h.p. peak performance would be unsuitable for bottom-end flexibility and economy. A working compromise has therefore to be accepted. With the rotary combustion chamber this is not so, as will be seen from a study of the performance curves. Broadly speaking, the larger the carburettor, i.e. as regards choke diameter and corresponding jet, the greater the maximum power without any appreciable set-off in other directions and without making any alterations or adjustments to the engine. In the Aspin designs there is in fact no "choke-critical".

Again, in conventional engines the economic reduction of the

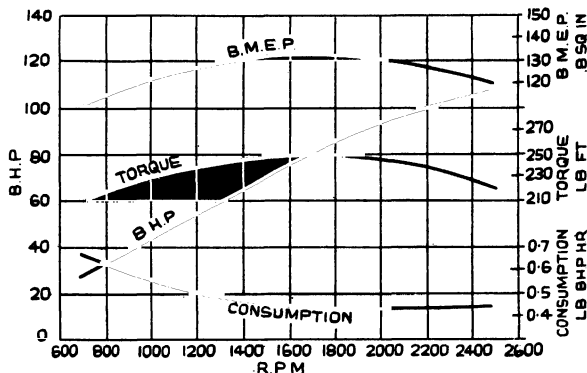



FIG. 90.
Performance curves of fully equipped 4·6 litre Aspin engine.

air/petrol ratio cannot be taken materially below 16 to 1 in a load curve test, i.e. full throttle and varying brake loads. Backfiring in the carburettor soon comes into evidence in most cases if attempts are made to use a leaner mixture than 16 to 1. With the Aspin ensemble this does not occur, and, especially at moderately high engine speeds, leaner mixtures can be used without any secondary ill effects. Air/petrol ratios leaner than 20 to 1 are quite satisfactory and economical running is obtained far above the 16 to 1 limit. The carburettor employed on this engine is a Solex type 46RZINE, the choke No. 38 and the main setting 250/250, which will be recognized as too large for a normal 4·6 litre swept capacity from which a reasonable performance at the lower and medium speed ranges is required.

The curves in Fig. 90 show how satisfactory are the characteristics with this carburettor over the entire range, and yet the diameter of the inlet manifold is 52 mm., which is on the large size for an

engine of this capacity and is not likely to produce much local turbulence, but on the contrary would tend to aggravate carburation troubles.

The reason for these remarkable results are, according to the Aspin theory, due to the whole of the cylinder contents being projected with considerable velocity into the cell from the confines of the narrow clearance above the piston, as discussed in the foregoing references to Fig. 85. Confirmatory evidence of the correctness of this theory would appear to be given by the fact that wide variations of choke and manifold diameters can be made without impairing the bottom-end performance. In other words, the design of the Aspin head, according to the hypothesis enunciated, looks after its own local turbulence down to the lowest speeds, the carburettor in no wise being required to perform an extraneous function so that it can be tuned to furnish a maximum top h.p without compromise in the matter of choke and jet diameters.

IGNITION TIMING. Nothing out of the ordinary is required in connection with the ignition timing. The great range of advancement generally necessary for ultra high speeds in poppet-valve engines is not required. The engine is singularly insensitive to any such need for reasons which are not yet fully explained, but are tentatively attributed to the centrifugal action to which both the live charge and the products of combustion are subjected.

PERFORMANCE. The curves in Fig. 90 give performance figures for an engine completely equipped with fan, water pump, dynamo, etc. An analysis of these curves shows several points of interest. The first is the unusual relation of the torque and fuel consumption curves. Generally, indeed almost without exception, the consumption is a fair inversion of the torque. At the lower range of speed the torque is low and the consumption high because of the combined effects of slow charge movements, uncertain distribution owing to poor suspension and indifferent combustion. This is common to all carburettor-supplied, plug-ignition engines with ordinary valve timings. Each reading then progressively improves up to the torque peak. Subsequently, the intrusion of wire drawing, with a corresponding drop in volumetric efficiency and therefore running compression ratio, gradually reverses this order, the torque progressively falling and the consumption rising. This rule is approximately correct for the Aspin engine in so far as the torque peak is concerned, but whilst the consumption behaves in the usual manner, and in the graph is a fair inversion of the torque, at the low speed end it flattens out after the inverted peak. This economy is maintained, as shown on Table 8, with only a fractional falling off, right to the end. By way of comparison, performance characteristics are shown for two well-known Diesel aircraft engines.

TABLE 8

PETROL CONSUMPTION OF ASPIN ENGINE

r.p.m.	*pints per hr.*	*pints per b.h.p./hr.*	*lb. per b.h.p./hr.*
1,500 ..	36·0	0·469	0·435
1,750 ..	41·5	0·464	0·428
2,000 ..	46·0	0·455	0·421
2,250 ..	49·5	0·454	0·420
2,500 ..	52·5	0·456	0·422

The Junkers Jumo 205-D German war engine (*see* Fig. 91) gives a fuel consumption of 0·35 lb. per h.p. per hour at cruising speed and 0·37 lb. at full load. Fig. 92 indicates the characteristics of the Guiberson A-1020 Diesel aircraft engine, of U.S.A. origin,* from which it will be seen that the fuel consumption at full load at 2150 r.p.m. was found to be 0·42 lb. per h.p. per hour, and at cruising

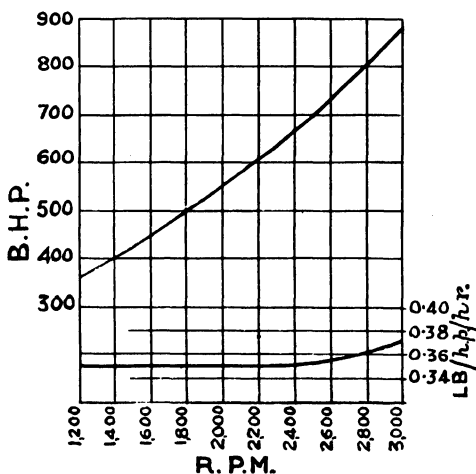



FIG. 91.
Fuel consumption of Junkers JUMO 205-D German war engine, Diesel principle.

speed the consumption was 0·37 lb. per h.p. per hour. It should be borne in mind, however, when comparing these consumption figures with the fuel consumption of a petrol engine, that Diesel fuel oil with a specific gravity of 0·840 weighs approximately

* *Diesel Aviation Engines*, Paul H. Wilkinson. New York. National Aeronautics Council, Inc., 1942.

7 lb. per gallon, whereas petrol weighs approximately 6 lb. per gallon.

The question may be asked how, then, in the Aspin engine, is this unusual consumption curve obtained? It would appear to be due to stratification of the charge in the eccentrically disposed rotating cell, as claimed by the designer. Higher b.m.e.p. than the maximum shown on the curve, namely 122 lb. per sq. in., have actually been recorded under different barometric conditions and with fuels of improved calorific value than those used for the particular tests recorded here. On brake test the engine will fire quite regularly and quietly on full throttle at under 500 r.p.m. This is remarkable for an engine with a compression ratio of 9·2

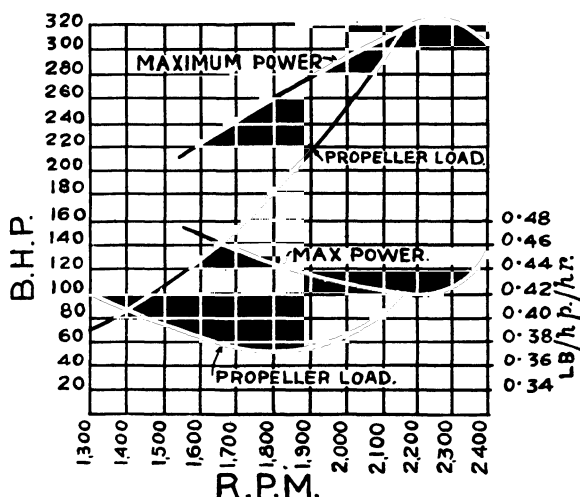



FIG. 92.
Power and fuel consumption of Guiberson A-1020 aircraft engine working on the Diesel principle.

to 1 and at the same time registering a b.m.e.p. of 100 lb. per sq. in.

The rotor, in its present developed form, has the ability of maintaining its condition and functions almost indefinitely. The designer offers a reasonable expectation of trouble-free operation for 100,000 vehicle miles, no servicing or decarbonizing being required, as the head keeps free of any appreciable carbon deposit. An engine is stated to have been run for half an hour at full load with the lubrication to the rotor completely cut off before any signs of imminent distress were observable. Even then, it squealed loudly for an appreciable time before suffering any serious injury.

PRESSURE PLATE TO REDUCE LOAD ON THE VALVE. A notable and fundamental improvement affecting the friction on the rotor

has recently been introduced into the design of the head, and this change considerably modifies the basic objection to the comparatively heavy loading on the conical surface in those cases where the Timkin thrust bearing has been eliminated from the construction.

The improved arrangement is shown in Fig. 93 and will be more easily recognized in Fig. 94, where it can be seen that part of the under-side of the rotor is masked off by a pressure plate formed by the cylinder top above the piston. The area of the rotor

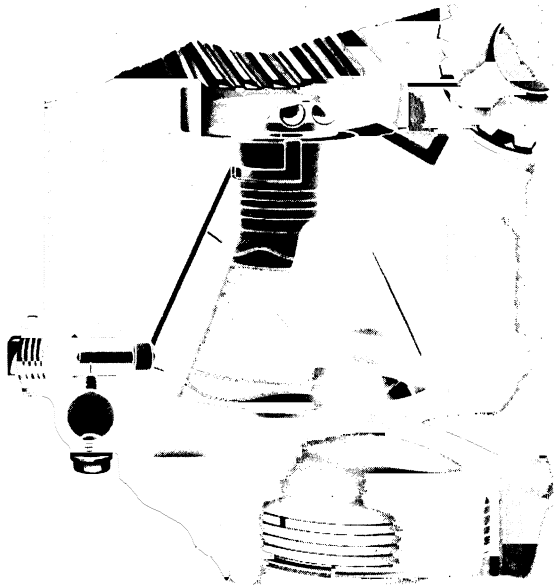



FIG. 93.
Aspin pressure plate construction with independent mounting. This view also shows the oil gallery and complete oil track.

which is then subject to the pressure within the cylinder is confined by a compression ring placed in a recess between the under-side of the rotor and the cylinder top. The diameter of the ring is proportioned to encircle the cell opening, and is located concentrically with the rotor, but not necessarily on the axis of the cylinder. The area bounded by the compression ring is approximately 0·40 of the area of the cylinder, so by this artifice alone the power-losses due to friction are reduced proportionately, a most substantial gain in the efficiency of a conical valve. The ultimate gain that is possible in a design using this artifice is, of course, limited by the required breathing size of the cell orifice, as the compression ring

has to be sufficiently large in diameter to encircle the cell opening to the cylinder. In computing the resulting load on the working surface of the rotor the angle of the taper is still to be taken into account, as discussed on page 133.

Referring again to Fig. 93 and Fig. 94, it will be seen that these illustrations, in addition to clarifying the principle upon which the pressure-plate device operates, indicate a further constructional feature which has been incorporated in the more recent designs. Each valve is contained in a circular box casting with

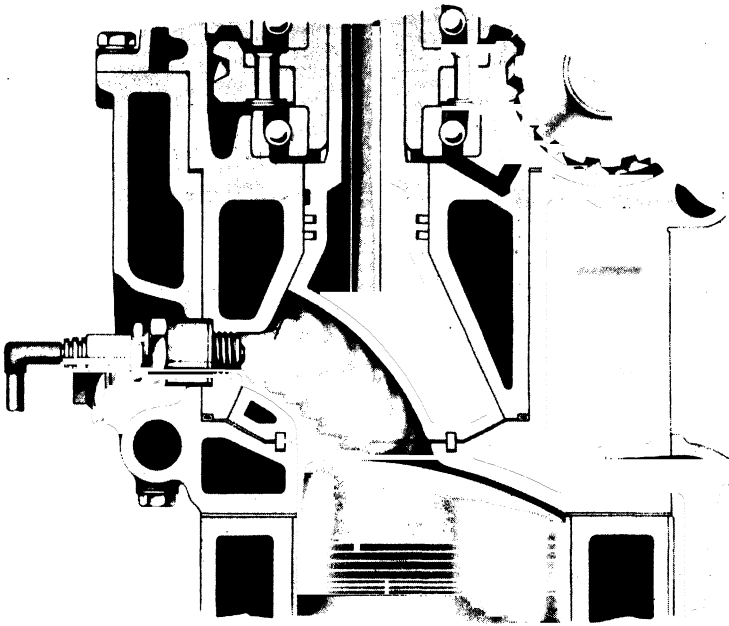



FIG. 94.
Aspin pressure plate construction, showing volume distribution at combustion.

its own water jacket. This enables any individual rotor to be removed and examined without dismantling the whole of the head of a multi-cylinder engine.

Hayes-Aspin Engine

The Hayes two-cylinder engine, designed by Mr. Aspin, has been primarily developed as a small power unit for agricultural purposes. It is also suitable for general industrial requirements.

The Hayes Engineering Co., Ltd., of Liverpool, are the manufacturers, and the company has adapted the engine to drive a small tractor designed for the purpose.

A general view of the engine on the test bed is shown in Fig. 95. The unit is an air-cooled vertical twin with a capacity of 1250 c.c., completely dustproof and is rated at 12 b.h.p. at 1500 r.p m. The two cylinders of $3\frac{1}{2}''$ bore with a stroke of $4''$ are cast en bloc with

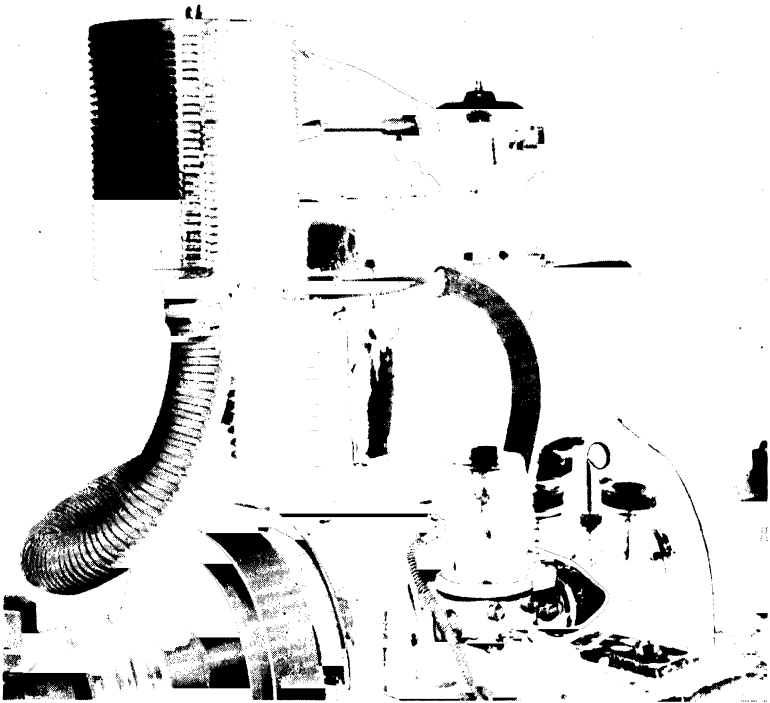



Fig. 95.
Hayes-Aspin two-cylinder general purpose 12 b.h.p. engine on the test bed.

the crankcase in ordinary cast iron. The crankshaft runs in roller bearings at each end with a plain bearing in the centre. The two inner crank webs maintain endwise location.

The two pistons rise and fall together, the crank throws being co-axial. This arrangement provides even firing at 360° and eliminates the objectionable couple associated with two-cylinder engines with cranks set at 180°. The primary balance is improved as far as possible by extensions to the crank webs, which act as balance weights. At its front end the crankshaft is extended and

supported in an outrigger roller bearing to carry the fan. This fan supplies cooling air to the cylinders through a cowling.

The design of the rotary valves follows Aspin practice and the rotors are housed in detachable inserts in extensions of the cylinder barrels. A bevel gear on the crankshaft drives a vertical spindle which at its upper end carries a pinion for driving the pair of rotors.

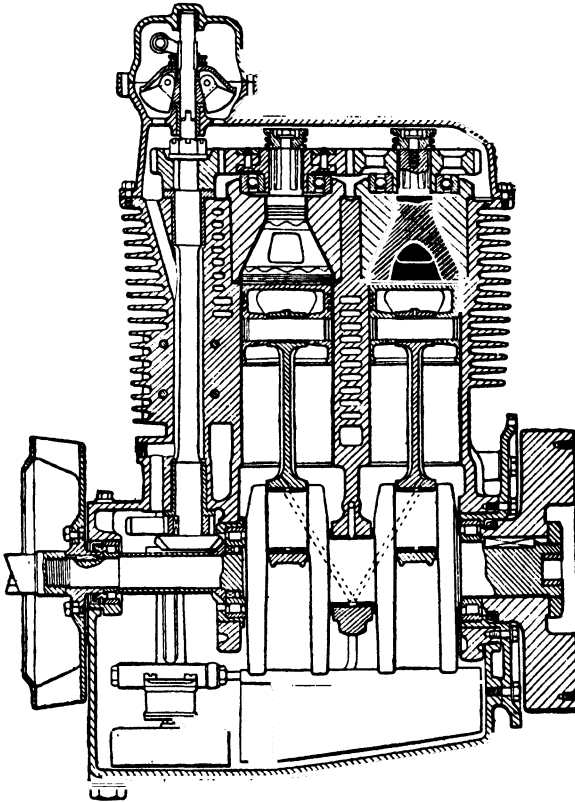



FIG. 96.
Sectional view of the Hayes-Aspin 12 b.h.p. general purpose engine showing governor and valve gear.

These are connected by straight-tooth spur gears of equal size. No idler between these gears is required and consequently the rotors revolve in opposite directions. Fig. 96 shows a vertical longitudinal section through the engine and valves. The governor will be noticed on the top of the vertical spindle that operates the valve gear.

The rotors are manufactured from close grain cast iron filled with an alloy of high thermal conductivity. The material used for

the detachable insert in which the rotor works is a special copper alloy with 25 per cent lead and 6 per cent tin. The port has an area of 1·72 sq. in., and as the bore of the cylinder has an area of 9·621 sq. in. the valve/piston area ratio is 0·177. The shape of the

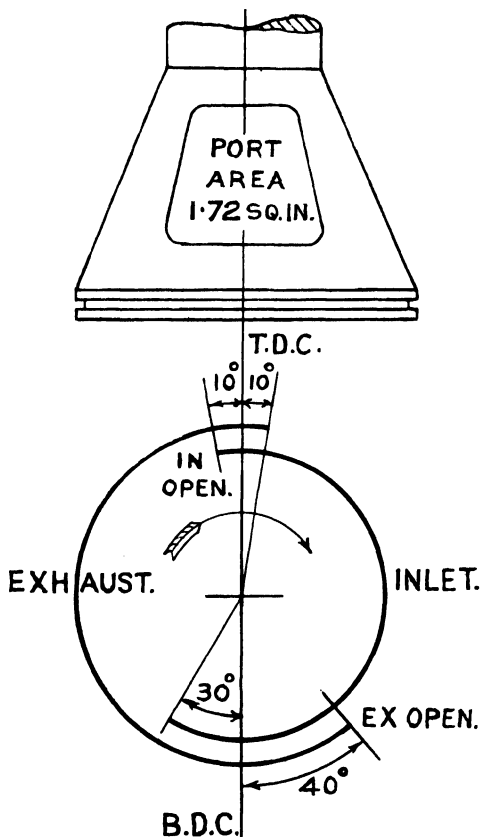



Fig. 97.
Shape of port and valve-timing diagram of Hayes-Aspin 12 b.h.p. engine.

port and the valve timing diagram is shown in Fig. 97. The overlap between inlet opening and exhaust closing being 20°.

A magneto supplies the spark for ignition and the point of firing is fixed, the spark occurring 5° before top-dead-centre.

The lubrication of the rotors is carried out on a system similar to that developed by Aspin for his 4·6 litre engine, i.e. the oil passing to the rotor is timed and also pressure controlled. The main oil supply under pressure is provided by a submerged pump

located in the engine sump and driven from the lower end of the same spindle which operates the rotary valves.

The fuel consumption, checked during a 100-hour test run on full load at 1500 r.p.m., is stated to be at the rate of 0·457 pint per b.h.p./hr. (approximately 0·43 lb.). The weight of the engine is approximately 2½ cwt. This may appear high, but it will be appreciated that weight is desirable for the purpose for which this engine was designed. At a speed of 2300 r.p.m. a b.h.p. of 20, without change in the carburettor settings, has been recorded.

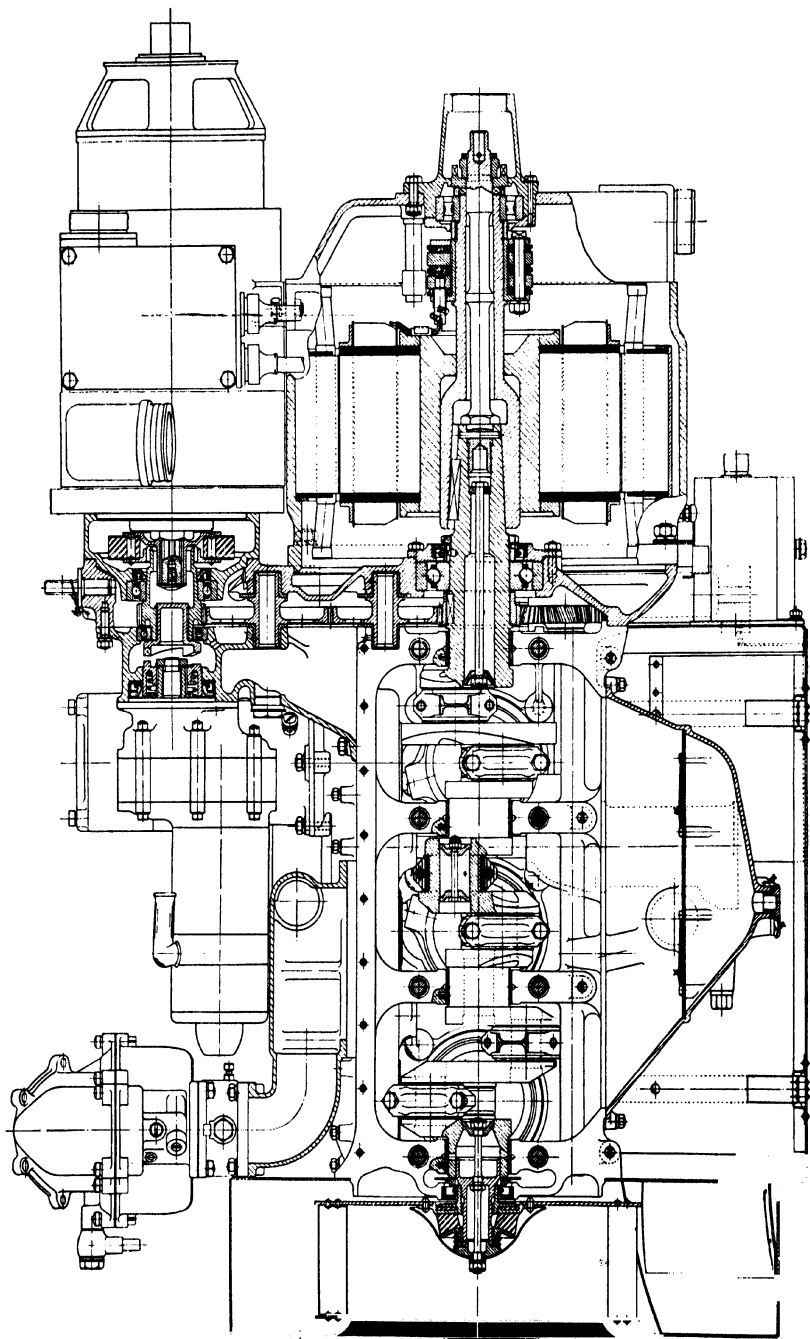



FIG. 98.
Longitudinal elevation of Rotol P6 equipment.

CHAPTER VIII

THE ROTOL AUXILIARY GENERATING PLANT FOR AIRCRAFT

THE Rotol Type P6 equipment provides electric current, both A.C. and D.C., for all the services on an aircraft, including starting the

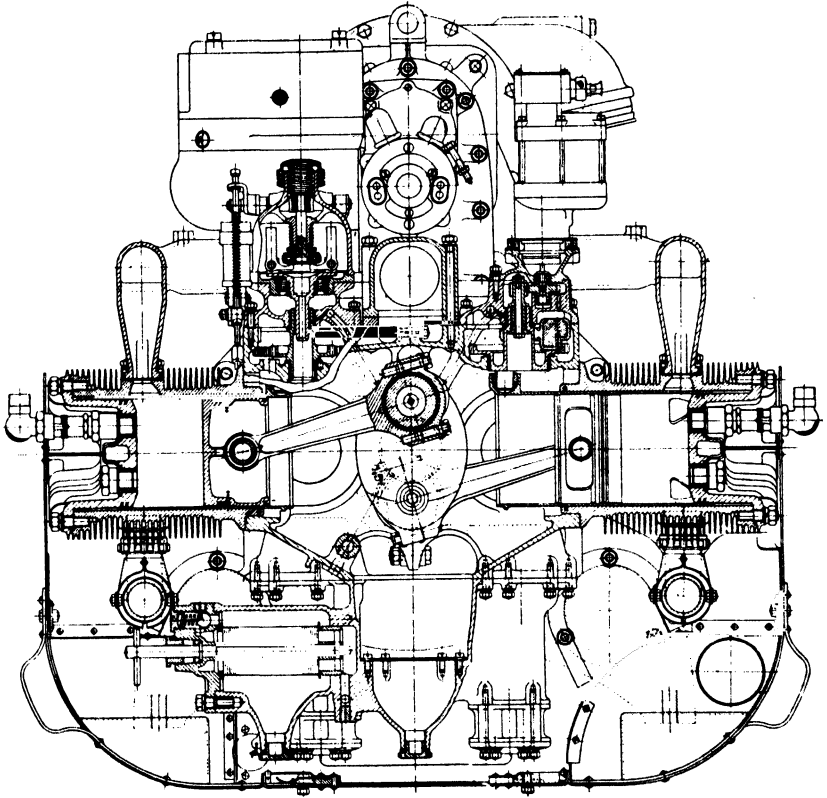



FIG. 99.
Cross sectional elevation of Rotol P6 equipment.

main engines. It consists of one or more independently driven electric generators, together with appropriate control gear. Longitudinal and cross-sectional elevations are shown on Fig. 98 and Fig. 99 respectively. To prevent any possibility of noise, fumes or vibration being transmitted to the body of the aircraft, the complete generating unit is housed in a sound- and fire-proof box. The sets are entirely automatic in action, governor controlled, and designed to

run for long periods without attention. The designers' choice of semi-rotary sleeve valves for such onerous duty indicates the great faith they have in the Burt principle.

General Description of the Plant

A six-cylinder horizontally opposed engine is directly coupled to an alternator. The revolving part of the alternator is mounted

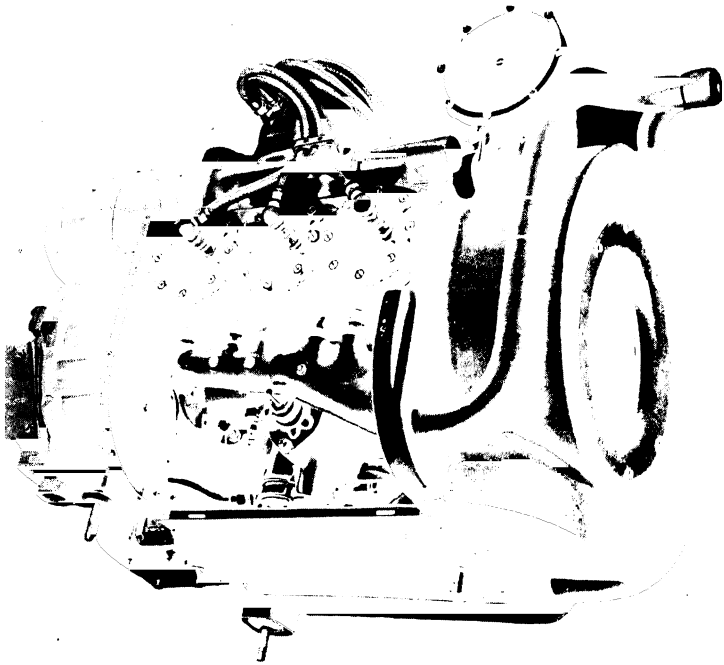



FIG. 100.
Rotol P6 equipment complete, without soundproof box and with side cover removed.

upon an extension of the engine crankshaft, and forms the flywheel for the engine. A gearcase is sandwiched between the engine crankcase and the alternator casing. This gearcase houses the engine timing gears and a train of gears driving the D.C. generator, and has two feet forming the main support of the unit. A third support is provided at the other end of the engine crankcase, giving three-point suspension.

A multi-blade centrifugal fan is mounted at the end of the crankshaft, and the system of air cooling is carried out in a most

efficient manner independently of the outside airstream, dissipating the heat from the generators, the exhaust pipes and cylinders with their junk heads. Great attention has been paid to the valve-gear mechanism, which is supplied with oil under pressure from the main oiling system of the engine. Fig. 100 shows the complete unit but without the soundproof box and with the side cover removed. The weight of the unit is 550 lb. including box, base and generators.

Main Features of the Engine

This compact engine is a first-class example of the application of semi-rotary sleeve valves to a modern air-cooled unit of medium horse-power. Although designed primarily for aircraft, the weight, dimensions and output are in line with contemporary engines designed specifically for use in road vehicles. The length of the engine without generators is approximately 24 in. There are six cylinders in flat formation with a bore of 3·375 in. and a stroke of 3 in., giving a total swept volume of 2·64 litres (161 cu. in.). The compression ratio of 7·5 to 1 is not as high as might be expected for a rotary valve, but it is higher than that normally used in poppet-valve engines. The nominal output is stated to be 60 b.h.p. at a governed speed of 3750 r.p.m. on continuous duty, and at a rated altitude of 12,000 ft. There is little doubt, however, that the stated output could be greatly exceeded for short periods if the engine was ungoverned and employed at altitudes nearer sea level as on ordinary motor-car work. It should be noted that the stroke bore ratio is less than square, being 0·8 to 1 in keeping with the latest trend in design for high rates of revolutions.

CONSTRUCTION. The six separate cylinders are machined from light alloy and bolted in a horizontal position to a two-piece alloy crankcase, three on each side. Opposing cranks are at 180°, and the cylinders are staggered as shown in plan view of the engine, Fig. 101. The inlet manifolds are on the top of the cylinders and the exhaust pipes below. The six-throw crankshaft is carried on four plain bearings and a large ball bearing located close to the alternator. The journals and crank pins, which are nitrogen-hardened, are hollow, and the crank pins are fed with oil from the journals through the usual drilled passages. A claim is made that the shaft is remarkably free from torsional periods. In addition the rubber mounting of the fan drive acts as a damper.

The connecting rods are of steel and have big-end caps secured by two set-screws, the joint face being serrated to ensure correct alignment. The small ends have bronze bushes. The light alloy

piston has two compression rings and two scraper rings, and a floating hollow gudgeon pin. The open ends of the ribbed cylinder barrels shown in Fig. 102 are closed by light alloy cylinder heads of conventional form, each carrying two compression rings for sealing purposes, and provision is made for two sparking plugs.

SLEEVE VALVE AND MECHANISM. The sleeve is the most important part of the valve gear, and successful operation depends upon

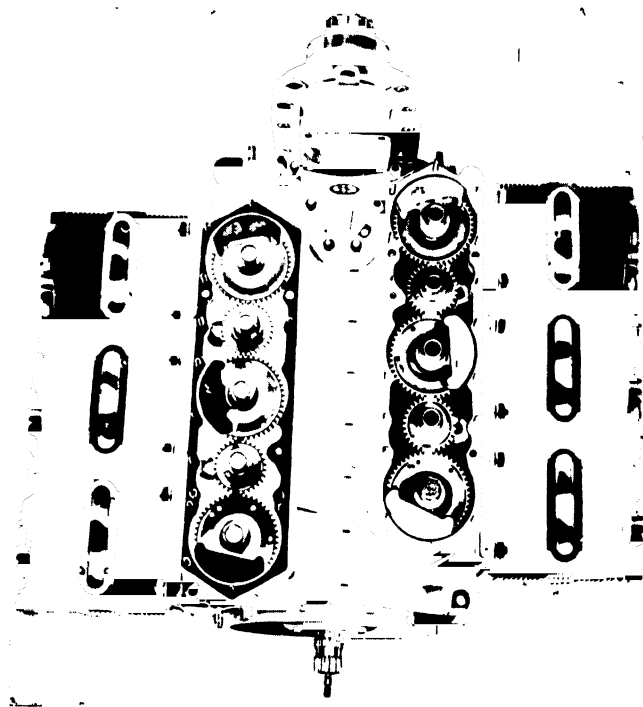



FIG. 101.
Plan view of Rotol six-cylinder horizontally opposed flat-six air-cooled engine with semi-rotary sleeve valves.

the accuracy of the finished component. Not only must the clearances specified be accurately held, but the surface finish both internal and external must be carefully controlled. Many grinding operations and several heat treatments for the removal of internal stresses are necessary to bring, in gradual steps, the internal and external diameters to within the final tolerances, limits of ovality and concentricity.

The material for the sleeve is a special steel, and any distortion arising from the nitride hardening may have to be corrected by rolling the sleeve on centreless grinding machines. After these

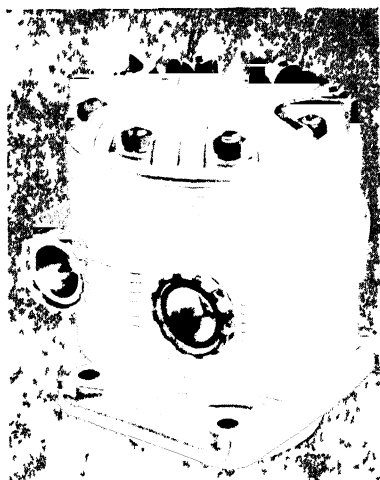



FIG. 102.
Rotol cylinder barrel showing the two exhaust ports and the light alloy head.

operations further grinding is required to bring the sleeve diameters within honing dimensions and a final brushing operation to impart a satin finish to the inner and outer surfaces.

The operating mechanism for the Rotol semi-rotary sleeve valve is shown in detail in Fig. 103, whilst Fig. 104 reproduces a

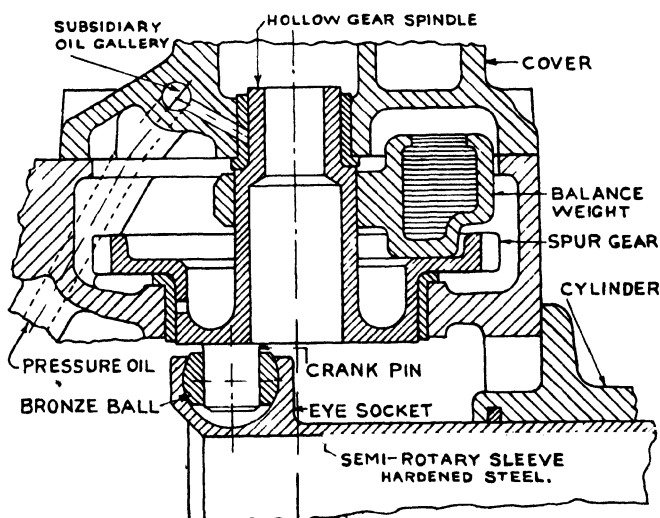



FIG. 103.
The operating mechanism for the semi-rotary sleeve valve of the Rotol engine.

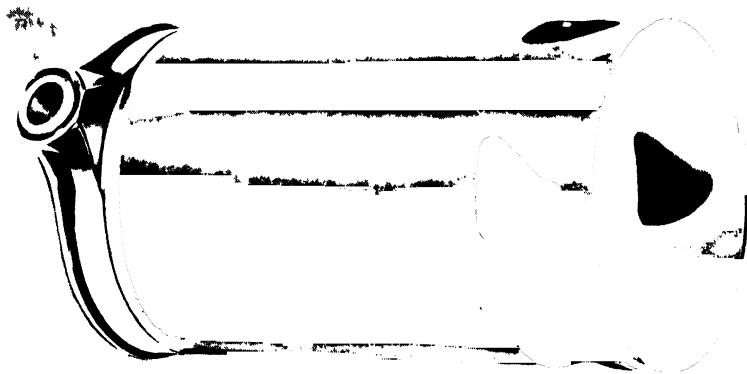



FIG. 104
The sleeve of the Rotol engine showing the four port openings, ribbed skirt and eye socket with bronze ball in position

photograph of a finished sleeve. It will be noticed that the crank pin is formed integral with the hollow gear spindle. The crank pin works in a bronze ball which floats in an eye socket machined from a solid lug on the skirt of the sleeve. The wall thickness is approximately 0.10 in. Each sleeve has four accurately

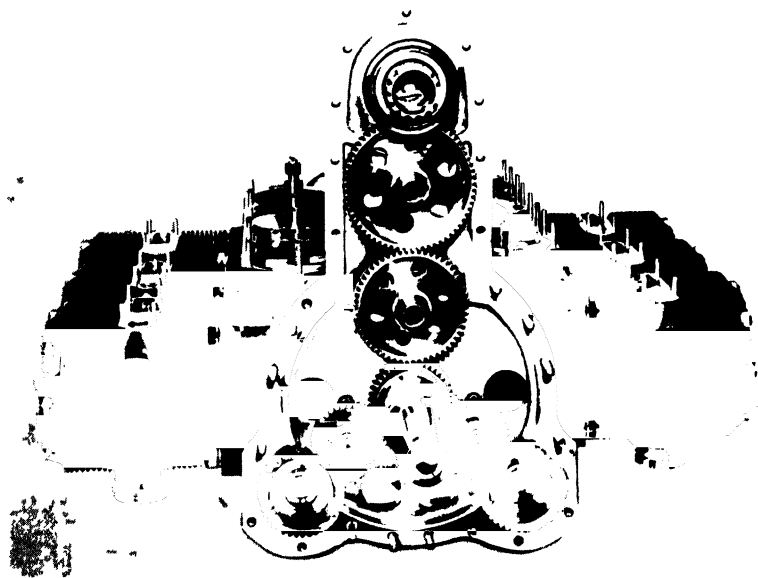



FIG. 105
Vertical train of gears on the Rotol engine driven from the pinion on the crankshaft

shaped port apertures working in conjunction with three inlet ports and two exhaust ports per cylinder. The valve timing is as follows:

Inlet opens 20° before T.D.C.
Inlet closes 50° after B.D.C.
Exhaust opens 60° before B.D.C.
Exhaust closes 10° after T.D.C.

The sleeve-operating cranks are mounted vertically on top of the crankcase and are connected together by a train of spur gears from each bank of cylinders. Small balance weights are attached to the gear wheels for the purpose of balancing the crank arm (*see* Fig. 101). Vertical spindles on either side drive the horizontal train of gears and also at their lower ends operate the oil pumps in the crankcase sump. These vertical shafts are driven through bevel gears by short horizontal spindles engaging with a train of helical spur gears located at the generator end of the crankcase and driven by a pinion mounted on the crankshaft. This pinion with the vertical train of gears is shown in Fig. 105.

The lubrication of the crank-operating gear and its bearings is provided by the main pressure oiling system of the engine. The subsidiary oil gallery feeding the valve crank mechanism and the drilled oil holes are clearly shown in Fig. 103.

Lubrication System

A pressure pump in the sump draws oil through a flexible pipe from the oil tank and delivers it under pressure to a self-cleaning filter built into the sump body. A safety valve is provided which lifts in the event of the filter becoming choked and permits the oil to by-pass the filter. A spring-loaded ball relief valve is built in the side of the filter casing. The spring is not adjustable and is set to control the oil delivery pressure between 40 and 50 lb. per sq. in.

Drilled passages conduct the pressure oil to a gallery running the length of the crankcase, whence passages are drilled to the four main bearings, and to the subsidiary gallery running along the top of the two sleeve-crank gear covers. The upper bearing bushes of the sleeve cranks and idler wheels are thus lubricated under pressure. Oil escaping from these bearings falls into the hollowed-out sleeve cranks, whence it is flung by centrifugal force into the sleeve crank lower bush and into the eye socket of the sleeve. The plain bearings of the spindles in the gear train between the sleeve cranks and the crankshaft are supplied with oil under

pressure by holes in the crankcase communicating with the main oil gallery.

After doing its work the oil falls into the sump below the crankcase, passing through a coarse gauge tray. The scavenge pump built into the sump collects the oil and returns it to the tank via the oil cooler. A by-pass is built into the cooler so that it is short-circuited until the oil entering the engine reaches its working temperature. The oil tank contains sufficient oil for 24 hours' continuous operation. The consumption is stated to be 0·80 pint per hour, and under normal operating conditions the temperature of the oil does not exceed 90°C.

Ignition

Ignition is provided by a screened six-cylinder magneto mounted vertically and driven by one of the sleeve cranks. An automatic timing device is incorporated in the magneto, which retards the spark for starting and advances the ignition progressively until full advance is reached at 2000 r.p.m. The firing point when fully advanced is 28° before T.D.C., measured on the crankshaft. A second drive is provided for another magneto or a coil ignition head, so that dual ignition may be fitted if desired.

Carburettor

Carburation is provided by a single down-draught carburettor centrally disposed and feeding branch pipes to the six cylinders. The carburettor has a built-in automatic altitude control, capable of satisfactory control up to 25,000 ft. A compact but powerful centrifugal governor operates the throttle and controls the engine speed within small limits from no load to full load. It is mounted vertically and driven by one of the vertical timing gear shafts. The engine runs efficiently on fuel of between 87 and 100 octane value. The consumption at normal rating is 3·5 gallons per hour.

Cooling System and Soundproof Box

The box has two horizontal duct openings at the engine end; one on the crankshaft centre and the other at the top of the box. The multi-vane fan (*see* Fig. 100), running at crankshaft speed, draws cooling air through the lower duct and discharges it into a box-like casing formed underneath the engine. Readily detachable panels give access to the engine sump and pumps. It will be

appreciated that the cooling air has four tasks: to cool the engine cylinders, the oil cooler, the alternator and the D.C. generator. The pressure in the duct forces the air to flow evenly around all of the cylinder barrels; conventional flat baffles on the upper or lee side of each bank of cylinders force the air to scrub the cylinders effectively. The heated air then escapes upwards into the roof of the box.

Exhaust manifolds of "aluminized" sheet steel, located inside the engine air duct, carry the exhaust gases to the duct end of the box. To protect the cylinders from radiated heat the exhaust manifolds are surrounded by sheet aluminium shrouds. These are suitably perforated, so that some of the duct air is forced along the shrouds and emerges into the box at the duct end, thus maintaining the manifolds at a satisfactory temperature.

Under ordinary running conditions the temperature of the engine does not exceed 160°C. A thermo-couple on the cylinder operates a temperature indicator located on the engine control panel.

CHAPTER IX

GERMAN DISK-VALVE ENGINES

Junkers Torpedo Engine. Model JUMO, KM8

GERMANY has always quickly seized the opportunity to investigate a new principle, to prove its worth or failings, and to carry out intensive development work on it to the smallest detail. It is therefore not surprising that between the years 1937 and 1945 a number of experimental engines, some twenty or thirty, with a new form of rotary valve had been built and tested and that when the World War came to an end a production order had been issued for the manufacture of 100 engines of an entirely new type, JUMO KM8, designed for the special purpose of propelling torpedoes.

The order for this batch of engines was never completed, but the prototype has been examined by British and American intelligence engineers,* and they have stated that it represents a progressive trend in general automotive development.

Full drawings and test data are not available, inasmuch as they were said to have been destroyed or evacuated from Dessau, where the experimental engine was made. However, an illustrated description was prepared by Dipl. Ing. Strohle, project engineer on the engine, and a translation provides a basis for an accurate specification.

High output, coupled with light weight, in the smallest package space is an important objective in torpedo design, and these requirements are equally important in all vehicle design. The KM8 engine was intended for a very short life under abnormal operating conditions, but it is thought that with some small changes in design, and operating under less onerous conditions, a longer life could be expected.

CONDITIONS TO BE FULFILLED. The ideas behind the design and the special conditions which the KM8 engine had to fulfil were:

(*a*) The engine to be capable of driving the torpedo at 40 knots, and for this purpose an output of 275 h.p. at 3650 r.p.m. was required.

(*b*) Minimum weight was essential, and therefore there was a strong leaning towards Junkers standard aircraft practice in design.

* *C.I.O.S. Report.* C. A. Lindblom, U.S. Ord., and Major J. R. Parry, British M.O.S.

(*c*) The engine had to reach full load and maximum speed in the shortest possible time. Starting from rest to running on a mixture of exhaust gas, oxygen and petrol had to be accomplished automatically following the throwing over of a single control lever.

(*d*) The engine had to be capable of withstanding the high temperatures and pressures resulting from the use of an exhaust gas and oxygen mixture.

(*e*) The engine had to reach maximum speed and power within one to two seconds from cold (starting being by a compressed-air starter motor). This entailed a very strong crankshaft, connecting rods, bearings, etc.

(*f*) The sparking plugs had to be so arranged that they were easily accessible from the outside of the torpedo with a minimum of inspection covers in the hull.

(*g*) The required life of the engine was only a few hours, so that a large wear and tear of individual parts could be tolerated.

(*h*) Surface treatment of materials and special materials to withstand oxygen and sea water had to be considered.

(*i*) Owing to the small available space in the torpedo, which was only 53 cm. internal diameter, the overall cross section of the engine had to come within this dimension.

As the planned capacity of the engine was 4·3 litres it was found impossible to incorporate normal overhead valve gear in the available space. Under these circumstances a rotating disk type of valve based on experimental work carried out by Wankel Entwicklungs Werke at Lindau, on Lake Constance, was employed. The overhead space required for this type of valve was approximately 45 mm.

DESCRIPTION OF THE KM8 DISK-VALVE TORPEDO ENGINE. The layout of the engine is shown in Figs. 106 and 107, being

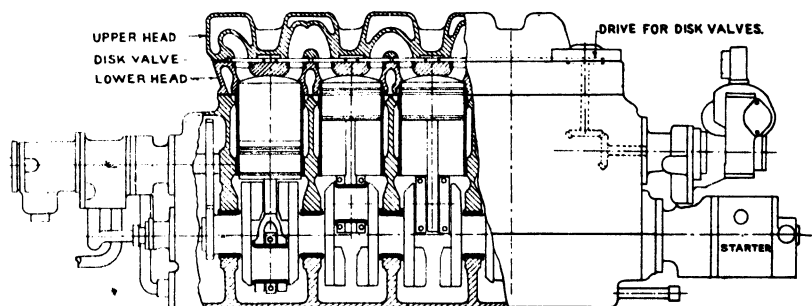



FIG. 106.

Longitudinal section through one bank of cylinders of German KM8 disk-valve engine.

longitudinal section and cross-section respectively. The clearance space between the outside of the various components of the engine and the inside diameter of the torpedo is seen in Fig. 107. The cylinder head with the rotary valve in greater detail is shown in Fig. 108.

This is a liquid-cooled internal combustion engine with magneto ignition, working on the four-stroke principle, and burning a

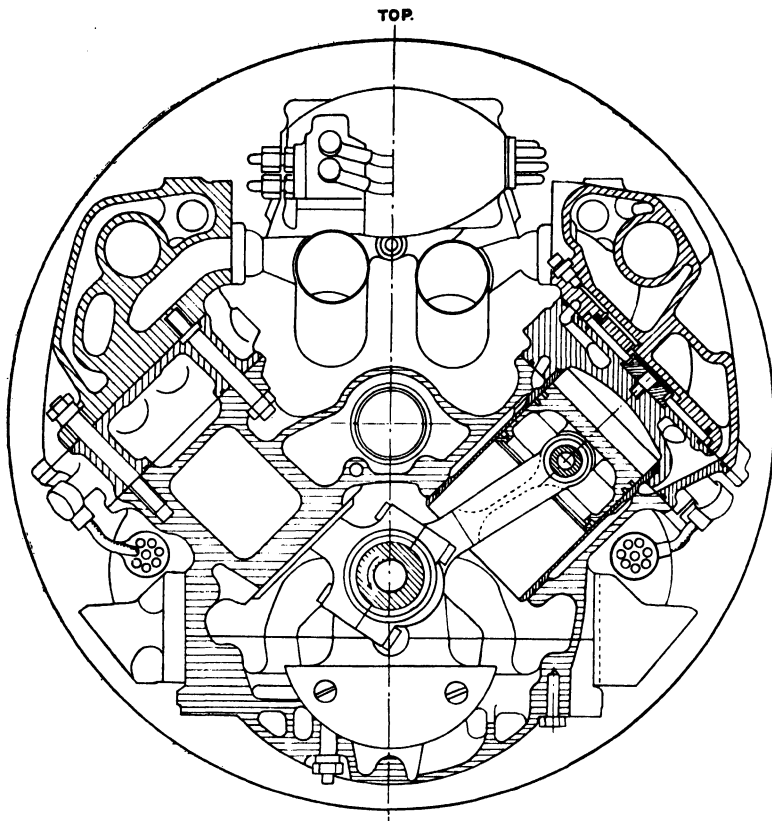



FIG. 107.
Cross section of German KM8 disk-valve engine encircled by the torpedo shell.

mixture of oxygen, exhaust gas and fuel. There are eight cylinders of 90 mm. bore by 85 mm. stroke, arranged in two banks of four in V formation at 90°, giving a swept volume of 0·542 litres per cylinder and a total swept volume of 4·34 litres for the whole engine. The compression ratio is 6·6 to 1, which is not high in comparison with some British rotary-valve engines.

The crankshaft is a nitrided steel forging with flanges to which the balance weights are bolted, and the main and the crank pin

journals are ground. Six main bearings are provided, the third from the front taking the end thrust. The connecting rods are H section steel forgings machined all over and fitted with split bearing shells of leaded bronze. To obtain greater strength the split in the bearing shell does not lie in the same plane as the split in the rod. Forged pistons of 12 per cent silicon light alloy are each provided with two compression rings and one scraper ring of conventional form.

The cylinder heads are cast in 9 to 10 per cent silicon light alloy to the same specification as the integral cylinder blocks and crankcase. Each head is divided into an upper and lower section to accommodate and to permit assembling the flat disk valves. The two heads are attached to the cylinder block by means of studs which pass through both sections. The lower cylinder heads are provided with water passages to allow the coolant, sea water, to flow from the cylinder blocks to the upper head to ensure good cooling to all high temperature surfaces adjacent to the valve chambers. Cast into the upper cylinder heads are inlet and exhaust ports corresponding with those in the lower half. The cylinder barrels are made from seamless steel tube. These barrels are screwed into the under-side of the lower half of the cylinder head, thus making a joint which requires no special sealing arrangement.

The system of lubrication for the engine is on more or less conventional lines, but the disk valves are lubricated by a separate multi-piston pump which delivers a fixed quantity of oil to each disk and to each disk bearing.

The rear of the crankcase housing accommodates the main reduction gear, the drives to the disk valves, with power take-offs for the rear oil pump and water pump, as well as the magneto, and pump lubricating the disk valves. The drive to the disk valves is not unduly complicated for an engine with cylinders in V formation.

The valve timing specification, as given below, is exceptional even for a supercharged high-speed engine, but the settings were based on considerable development work on single-cylinder test engines. The long periods and large overlap are necessary to suit the comparatively small port areas. Valve timing:

Inlet opens 40° before T.D.C.
Inlet closes 55° after B.D.C.
Exhaust opens 71° before B.D.C.
Exhaust closes 24° after T.D.C.

THE DVL WANKEL TYPE ROTARY VALVE. The outstanding feature in the specification of the engine is the disk-valve gear, which

takes up much less space than the conventional overhead poppet-valve, and the components are simple to manufacture. The disks

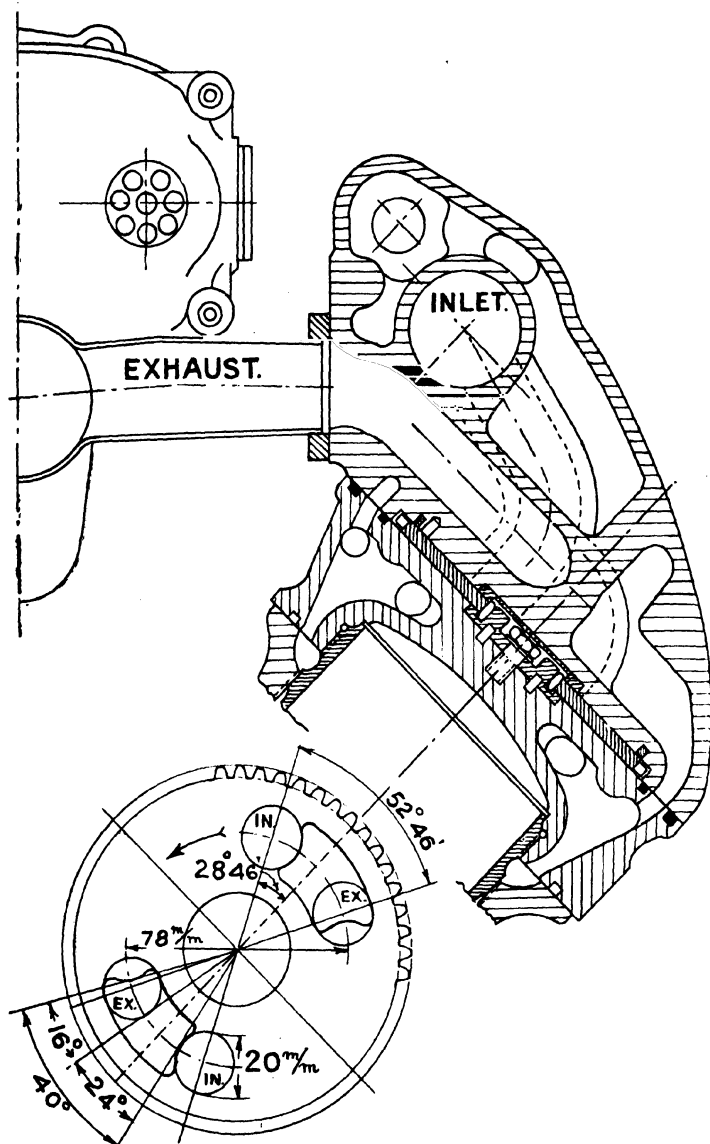



Fig. 108.

Cylinder head of German KM8 engine, showing the arrangement of the rotary disk valve, the ports in the cylinder head and the shape of the apertures in the disk.

are provided with teeth cut on their peripheries and form two gear trains without the need of gear wheels or idlers.

The valve disks are fitted between the upper and lower cylinder heads, the upper head being machined out to provide suitable clearance. Two inlet and two exhaust ports per cylinder are cast into the lower cylinder head and lead to the under-side of each disk. All the ports are circular and each is fitted with a steel sealing insert which is held against the disk by the pressure in the cylinder, thus effecting a gastight joint. The outside diameter of the steel sealing insert is made gastight by two ordinary piston-type compression rings. The upper side of each lower cylinder head is machined out above the centre of each cylinder to accommodate the valve bearings, which are pegged to prevent rotation. A set of needle rollers is fitted between the inner diameter of the disk and the outer diameter of the bearing to reduce friction.

It is important to note that the opening through the sealing insert is circular and the section of the material sufficient only to accommodate the compression rings, consequently the area under pressure and the resulting friction are a minimum.

There are two port apertures in each disk as shown in Fig. 108, and they are shaped to provide a quick rate of opening and a quick cut-off. This unusual shape is necessary because the hole through the sealing insert is truly circular. The figure also shows the overlap and the correct relationship between the exhaust and inlet ports to meet the timing specification. The ports are all the same diameter, namely 20 mm., and as there are two inlet ports per cylinder this gives a valve/piston area ratio of approximately 0·1. The valve area, however, appears to be sufficient in view of the high horse-power attained on test. The gas pressures acting on the disk valves are counterbalanced by means of thrust bearing rings fitted into the upper cylinder head.

It will be appreciated that the success or otherwise of this type of valve depends to a large extent on the accuracy of the machine work and also upon the variation in clearances as a result of working tolerances. Great attention has been paid to this matter, and the manufacturing dimensions, with the important maximum and minimum clearances, are given under Fig. 109.

The train of disks for each bank of cylinders is driven from pinions situated at the rear of the engine and by bevel gears from the crankshaft. The spindles driving the disk valves are provided with splined couplings which allow exact timing of the valves to be obtained. Shear pins are fitted to the gears driving the disk-valve trains to prevent damage of the disks and gears should the disks seize. The gear ratio between the revolutions of the crankshaft and the disk valves is 4 to 1.

Great importance is attached to the material specifications

for the various rubbing parts of the valve gear. The steel for the port sealing inserts is to the following specification: C 0·33 to 0·41, Si 1·1 to 1·4, Mn 1·1 to 1·4, P 0·035 max., S 0·035 max. Remainder Fe. The material for the compression rings for the sealing inserts is a cast iron of a grade usually used for engine piston rings. A nitrided steel, of the same specification as used for the crankshaft, was employed for the disk valves, the analysis being: C 0·26 to 0·34, Si 0·4 max., Mn 0·4 to 0·8, P 0·035 max., S 0·035 max., Cr 2·2 to 2·5, V 0·15 to 0·25. Remainder Fe.

TEST RESULTS. The output of 275 h.p. at 3650 r.p.m. was obtained when running on a mixture of oxygen and exhaust gas

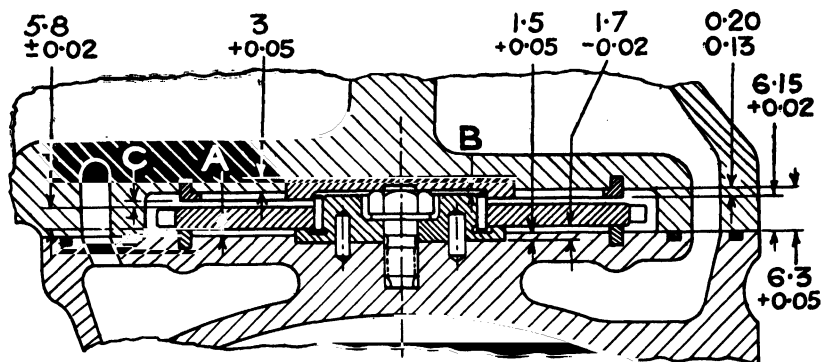



FIG. 109.
Dimensions and manufacturing tolerances for the DVL Wankel type rotary disk valve on German KM8 engine. Dimensions given in millimetres.

	Max.	Min.
Clearance at A.	1·7 − 1·5 = 0·2	1·68 − 1·55 = 0·13.
„ „ B.	6·35 + 3·05 + 1·55 = 10·95 and 10·95 − (8·7 + 1·9) = 0·35	6·3 + 3 + 1·5 = 10·8 and 10·8 − (8·8 + 2) = 0.
„ „ C.	6·17 − 0·13 = 6·04 and 6·04 − 5·78 = 0·26	6·15 − 0·2 = 5·95 and 5·95 − 5·82 = 0·13.

with a supercharge pressure of 1·5 atmospheres absolute. It was necessary to increase the pressure to 2·5 atmospheres in order to gain the maximum of 425 h.p. at 4360 r.p.m. Running the engine as a normal petrol engine, without supercharge, but using the same simple carburettor and induction pipe as was used when running on exhaust gas and oxygen, an output of 110 to 120 h.p. was obtained.

There were no signs of pre-ignition, even at maximum output, because the mixture of exhaust gas and oxygen on which this engine runs contains a high percentage of water.

When running on the oxygen mixture, difficulties were experienced with the bearings of the disk valve and the gas tightness of the sealing inserts. The plain bearings originally fitted were

not satisfactory owing to lubrication difficulties, and these were replaced with needle roller bearings which gave very good results.

Owing to the very high temperatures, there was a tendency of the port inserts to burn. After the material of these inserts and that of their compression rings had been changed to the specifications already given, this trouble was largely eliminated. When running the engine on "free air" practically no sealing difficulties were encountered. The engine has been run as long as fifty hours on "free air" without any trouble. (A Junkers 210-S single-cylinder engine has a similar disk-valve gear and has run 200 hours and more, trouble free.)

During the tests another change of design was necessary to avoid wear due to the disks bearing too heavily against the lower surface of the upper cylinder head as a result of the high explosion pressures acting on them. Two bronze thrust rings were let into the upper cylinder head with provision for pressure lubrication.

By the methods outlined above the troubles experienced with the original design were eliminated. As a precautionary measure the bearings of the disk valves have been strengthened on the latest engine by increasing the thickness of the disks. As the design of this engine is to a large extent based on experimental work carried out on larger single-cylinder engines of 124 mm. bore, the report of these tests is given in full. The information provides important facts regarding friction horse-power and friction M.E.P. compared with a poppet-valve engine of equal size.

Report of Tests on Experimental Single-Cylinder Engines

Comparative tests of an exhaustive nature were carried out on single-cylinder engines fitted with poppet-valves and DVL disk valves. In addition to the actual behaviour of the disk valve, it was particularly desired to obtain a comparison between this type of valve and the JUMO three-valve arrangement. With this in view the following tests were carried out:

A. Output data: Power consumption and intake of air at various speeds and supercharging pressures.
B. Limits of power: (Borderline of knock).
C. Running data: Maximum pressures at various loads.
D. Peculiarities of operation.

TEST APPARATUS.

Tests were carried out on two single-cylinder engines. In order to obtain strictly comparable results, stress was laid on obtaining a similar layout in the two test engines.

Constructions: 1. Poppet-valve engine E210F.
2. DVL disk-valve engine E210S.

124-mm. bore, 136-mm. stroke } 1·67 litre swept volume.

During the course of the test the valve timing of the 210F, i.e.

Inlet opens 15° before T.D.C., closes 52° after B.D.C.
Exhaust opens 55° before B.D.C., closes 42° after T.D.C.

was improved in order to attain high speeds.

Tests were run on the disk-valve engine with various port openings based on information obtained from D.V.L., as well as to determine the corresponding injection and ignition timings for optimum results. The comparison tests were then run with the valve and ignition timings listed below:

E210F—three-valve (poppet-type) engine:
Compression ratio = 6·5 to 1.
Commencement of fuel injection 60° after T.D.C.
Ignition 40° before T.D.C.
Inlet opens 35° before T.D.C.
Inlet closes 50° after B.D.C.
Exhaust opens 70° before B.D.C.
Exhaust closes 44° after T.D.C.

E210S—disk-valve engine:
Compression ratio 7·5 to 1.
Commencement of injection 90° after T.D.C.
Ignition 30° before T.D.C.
Inlet opens 37° before T.D.C.
Inlet closes 73° after B.D.C.
Exhaust opens 89° before B.D.C.
Exhaust closes 21° after T.D.C.

Both engines were run with diametrically opposed sparking plugs.

A. *Power Data* (power, consumption and intake of air at various speeds and supercharging pressures):

The tests were run at speeds between 2000 and 3300 r.p.m., and with supercharging pressures of 1·03 atmos., 1·17 atmos. and 1·35 atmos. absolute. The following was determined:

1. The quantity of air used per stroke in grams/stroke (*see* Fig. 110).
2. The B.M.E.P. at the different supercharging pressures at various speeds (*see* Fig. 111).
3. The optimum fuel consumption in Gr./B.H.P./Hr. (*see* Fig. 112).

Fig. 113 shows the values obtained plotted as consumption hooks. The air/fuel ratios were not shown as a basis of comparison, as the engines possess different air consumption characteristics.

As the B.M.E.P.s of the two engines are only strictly com-

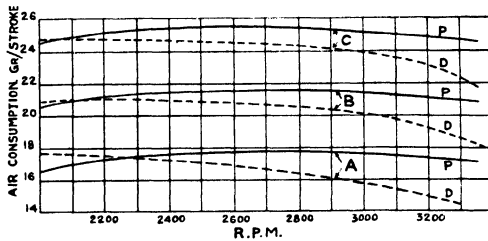



FIG. 110.

Comparison between poppet-valve and DVL disk-valve engine, air consumption at various supercharging pressures.

A. Supercharge at 1·03 atmos. absolute.
B. ” ” 1·17 ” ”
C. ” ” 1·35 ” ”
P. Poppet-valve engine.
D. Disk-valve Engine.

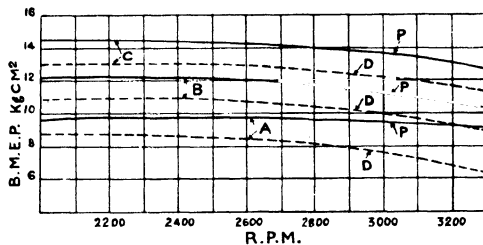



FIG. 111.

Comparison between poppet-valve and DVL disk-valve engine, brake mean effective pressure.

A. Supercharge at 1·03 atmos. absolute.
B. ” ” 1·17 ” ”
C. ” ” 1·35 ” ”
P. Poppet-valve engine.
D. Disk-valve engine.

parable when friction is taken into account, Fig. 114 shows the friction H.P. and F.M.E.P. of the two engines. These values were checked at intervals throughout the tests.

B. *Limits of Power* (Borderline of knock):

The presence of knock was recorded by a cathode ray oscillograph.

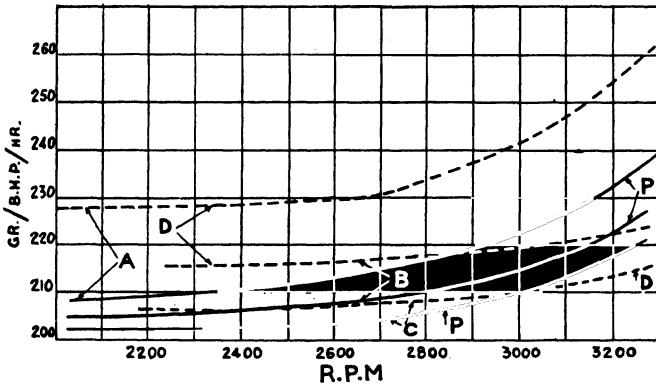



FIG. 112.
Comparison between poppet-valve and DVL disk-valve engine, optimum fuel consumption at various supercharge pressures.

A. Supercharge at 1·03 atmos. absolute.
B. „ „ 1·17 „ „
C. „ „ 1·35 „ „
P. Poppet-valve engine.
D. Disk-valve engine.

The limit of knock being taken as ten per minute, the amplitude being less than 70 mm. on the reference chart in use.

The border line of knock on the poppet-valve engine (ignition timing 40° before T.D.C., air temperature 30°C.) lies in the neighbourhood of 15·8 Kg./cm.² B.M.E.P.

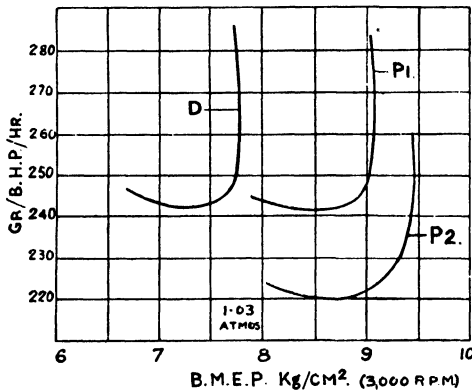



FIG. 113.
Comparison between various valve arrangements with open throttle. Variable mixture consumption hooks.

D = Disk valve.
P1 = Poppet-valve old timing.
P2 = „ „ new timing.

After running for long periods at this load no signs of knock could be detected. The engine was therefore run at 16·65 Kg./cm.² B.M.E.P. for two and three-quarter hours, which produced seventeen knocks per minute of amplitude 70 mm. on the reference chart. On completion of the run the *piston* showed slight traces of knock. The figure given for the limit of knock can therefore be considered as permissible.

The borderline of knock on the disk-valve engine (ignition timing 30 B.T.C., air temperature 30°C.) lies in the neighbourhood of 14·7 Kg./cm.² B.M.E.P. This engine was then run for two and three-quarter hours at 15·5 Kg./cm.² B.M.E.P., and on being

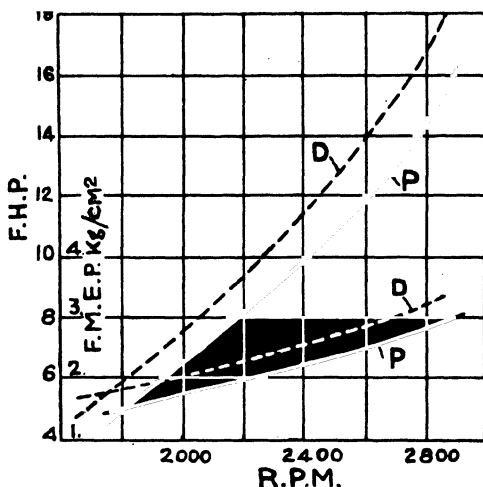



FIG. 114.
Comparison between poppet-valve and DVL disk-valve engine. Friction H.P. and friction M.E.P. Values taken from run-in engine and engine well warmed up. Water temp. 80°C. Oil temp. 65°C.

Supercharge at 1·03 atmos. absolute.
P = Poppet-valve engine.
D = Disk-valve engine.

stripped showed slight signs of knock at the *inlet-port* sealing insert. If the disk-valve engine is allowed to run under heavy knock conditions the port inserts become red hot (burning), which leads in a short time to self-ignition.

C. *Running Data* (Maximum pressures at various loads):

A basis for the working power of the engine was to be established by determining the maximum pressures. These were determined by means of an oscillograph, the quartz transmitter being

inserted into the inlet port. For mechanical reasons the transmitter had to be inserted into the inlet or exhaust ports. The difference in results obtained from the two positions was not established.

The maximum pressures varied as is usual, and for this reason 30–40 consecutive working strokes were recorded.

Fig. 115 shows the maximum pressure of the poppet-valve

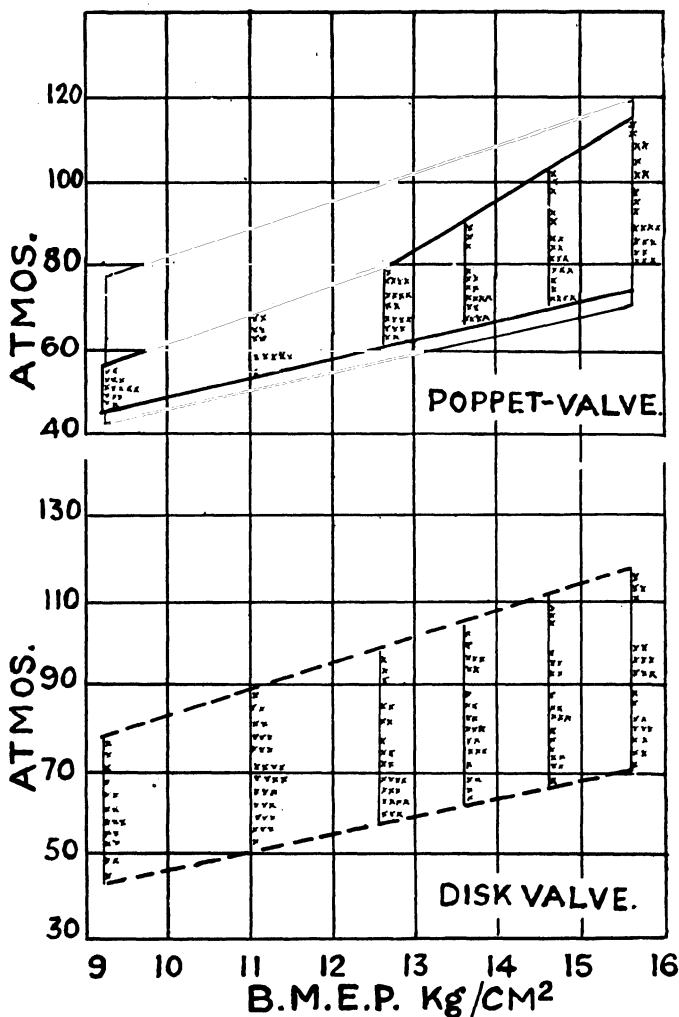



Fig. 115.

Comparison between poppet-valve and DVL disk-valve engine. Area of variation in maximum pressures. The frequency is given by points lying in the same horizontal line. B.M.E.P. through which lines are drawn were the only ones tested.

engine E210F on a B.M.E.P. basis. The compression ratio being 6·5 to 1, air intake temperature 30°C., cooling water temperature 80°C. and oil temperature 60°C.

Fig. 115 also shows the maximum pressures of the disk-valve engine E210S on a B.M.E.P. basis. The compression ratio is 7·5 to 1, the other conditions being the same as for the E210F engine.

D. *Peculiarities of Operation:*

The disk-valve engine ran satisfactorily at medium power output, but difficulties were encountered with heat dissipation from the port inserts when running at high M.E.P.s. With an M.E.P. of 15·5 Kg./cm.^{2} maximum self-ignition took place, the engine running with slightly decreasing power for a few minutes after the ignition was switched off.

At the end of the test run it was found that the port inserts and sealing rings were badly burned.

In order to determine the effectiveness of the sealing insert ring under load a ten-hour run was made with a weak combustion mixture.

Ignition 28° before T.D.C.
M.E.P. 12·3 to 12·5 Kg./cm.^{2}.
Cooling water temperature 80°C.
Oil temperature 60°C.
Supercharging pressure 1·42 atmos. absolute.

At the end of the run the disk valve was in good condition, but carbon was forming on the edges of the slots; the amount of oil delivered to the valve could be reduced without detriment.

The quantity of oil used by the disk valve during the test was 1·24 Gr./B.H.P./Hr.

The port sealing inserts and sealing rings were in good condition.

At the end of a 52-hour run it was found that the port insert sealing rings had formed lips on the port insert seat.

This results in jamming of the port inserts and imperfect sealing. (Material of the port insert seats Y alloy.)

CHAPTER X

CLAIMS FOR THE ROTARY VALVE, FUTURE TREND AND REFLECTIONS

THE numerous advantages of the rotary valve have been discussed in considerable detail and some of the claims are supported by eminent engineers. The chief advantages are emphasized in the following list.

Advantages of the Rotary Valve

(1) Rotary valves do not require periodic adjustment, and the opening and closing of the ports is positive at all speeds without maintenance.

(2) Their action is, or can be, noiseless, with no limit to speed. The modern poppet-valve engine is remarkably silent but gained only with some sacrifice of power.

(3) The valves require no attention in the way of regrinding and cannot readily be thrown out of tune by misuse.

(4) Their employment permits the use of combustion chambers of more uniform shape with the sparking plug situated near the centre, hence the indicated efficiency is high and the tendency to detonate at a minimum.

(5) Improved thermal stresses due to the symmetrical aspect of the cylinder head, which is less influenced by the valve.

(6) Perfect balance of rotors is possible in most designs. In the Burt semi-rotary sleeve, as the velocity is almost uniform, there are no appreciable acceleration forces, and so far as the valve gear is concerned the speed is almost unlimited.

(7) Absence of spring breakages and no tappet vibration.

(8) No exhaust valve head to form a hot spot.

(9) Superior breathing characteristics gained by streamline ports. Increased turbulence at low r.p.m. due to valve action with changing tangential flow of the incoming charge.

(10) Less frequent necessity for decarbonizing. At low power conditions a small carbon deposit may form in the angle of the ports, but on opening up to full power this is blown out.

(11) Very low rate of wear. In the case of the semi-rotary sleeve the wear inside the sleeve is considerably lower than that in the cylinder bore of a poppet-valve engine owing to the changing bearing surface brought about by the rotary movement.

(12) In some designs the intrinsic cost is less than the poppet-valve engine and the number of parts may be fewer.

(13) The compression ratio can be increased and fuel of low octane value used.

(14) The specific fuel consumption is lower.

The aggregate of all these claims results in greater horse-power for a given size of engine, and without any need for a high grade of fuel.

Octane Value of Future Fuel Supplies

The general design of automobile engines in the immediate future will depend to some extent on the commercial grade of fuel which is likely to be in easy supply. It has been widely assumed in some quarters that 100 octane petrol developed in such large quantities for war planes will be the super fuel of post-war automobiles, but this view is not expressed by Mr. D. P. Barnard,* Assistant Director of Research of the Standard Oil Co. of Indiana, who says, "Actually, the octane number of fuel generally available for automobiles in the future is not likely to average more than three numbers higher than that of the pre-war product." In these circumstances it is highly probable that great attention will be concentrated on the development of engines which will be able to use low octane spirit, and the rotary valve in its present stage of design already fulfils this condition to a remarkable degree. World demand for petrol of all grades in the future is likely to be tremendous, and the cost is not expected to fall much below its present level. For these reasons the remarkably low fuel consumption already attained by engines employing the rotary valve is a great incentive for further development of the type, especially if manufacturing costs can be established on a comparable basis with the poppet-valve engine.

Need for Improvements in Mechanical Efficiency

It is generally admitted that intrinsically the rotary-valve engine is not more expensive than the poppet-valve unit, but the ability to produce the latter cheaply is only the result of years of learning how to tool properly for large quantities. This fact is a definite handicap to the competitive manufacture of the newer valve, but it is a handicap which cannot be expected to persist indefinitely.

The practical problems of securing the full results from the use of the rotary valve are only partly solved, but the ability to operate consistently at high speeds and for periods of long duration has been well proved. This is basically due to the increased knowledge of materials, improvements in methods of manufacture,

* Vide *Commercial Car Journal*, U.S.A., April 1944.

together with the ability to produce work to closer tolerances and finish than was possible even a few years ago. In spite of the advances made in these respects there is still much investigation to be carried out in a number of directions.

It will be gathered from the full descriptions of up-to-date designs illustrated in previous chapters that the two most advanced manufacturers have adopted divergent methods of construction and entirely different systems for sealing, and it may be that much new ground will be broken in various fields before the foundations of stability in design are laid, but the marked advances that have been made during the past few years indicate that the evolution of the practically perfect rotary valve is not far distant.

The systems in most general use absorb a considerable amount of power and therefore generate heat which has to be dissipated. Power-losses must be reduced by more clever design to such a small order that the heat units, produced as a direct result of mechanical friction, and which add to the prevailing temperature resulting from combustion, are of negligible account. This is a precept which has been stressed very strongly throughout this book, but not too strongly, because the poppet-valve adds zero units of heat to the cylinder head.

On first considerations the rotary valve appears to be a simple mechanism, as in principle indeed it is, but the vision of one rotary member replacing a hundred or more components on a four- or six-cylinder poppet-valve power unit is far removed from actual facts. This has been fully demonstrated in the numerous examples of current designs fully worked out with sealing devices, obturators and oil scrapers or alternative systems of lubrication. The collection of parts in the complete assembly is then found to be hardly fewer than the number of components making up the more orthodox poppet-valve system.

However, at the present time this is not a world in which simplicity and fewness of parts are the only determining factors. Efficiency has to be gained at the expense of simplicity, if there is no other means in view to gain the desired results. The modern aero engine is only one typical example of this precept. Had the single-cylinder engine been retained on the score of simplicity there could have been little forward progress in the automobile industry as we know it today.

The general trend is likely to be a striving for greater efficiency and for new means of producing the components in a more economical manner. For these reasons it may be said there is small justification for adhering to, say, a design with a single rotor for intake and exhaust, if two separate revolving members of more elementary design can be made to give effect to improved performance at commensurate cost.

AUTHOR'S CONJECTURES

Reductions in the basic cost of materials are of frequent occurrence and variations in their relative values take place from time to time. For instance, aluminium castings made from secondary, i.e. remelted, aluminium can compete today in cost with iron castings, and the former material is likely to be used to a larger extent for cylinder heads and intake manifolds, and also for cylinder blocks, especially in the case of rotary-valve engines, where in general the block is of simple form.

In future lines of developments it is reasonable to expect some improvement in mechanical design, in metallurgical properties, methods of lubrication and sealing arrangements, in spite of the worthy results obtained at the present time by a few manufacturers. It will at the same time be readily admitted that the primary difficulties have been completely overcome. Some factors demanding attention may be summarized.

In the field of special materials there is still much research work ahead, more particularly in regard to the best mating properties of the two elements forming the basic friction surfaces—the rotor and the sealing component—this applies particularly to designs where actual contact is the principle employed to ensure a gas-tight joint.

On the other hand, the thermal characteristics of metals provide good ground for exploration, the expansion coefficient being of paramount importance when a fine annular clearance between rotor and stator is aimed at in the design and consequently relied upon to a greater extent for sealing. This principle has been tried at various times but has been generally abandoned. In order to preserve such a fine dimensional clearance, as is necessary to meet all variations of temperature as well as all conditions of speed and load, there is a call for deeper investigation, together with exhaustive experiments to prove not only the best combination of materials, but also the matched thermal characteristics to suit a specific design of valve and head. It may be that this principle tables a problem outside the bounds of practical solution.

The thermal conductivity of a material is also a most important physical property, offering scope for further research. Aluminium, with its high value of conductivity, approximately four times that of cast iron, naturally serves as a useful vehicle for the transfer of heat from the valve and combustion head to the cooling medium, but it has only about half the conductivity of copper, the comparison being 0·05 for aluminium and 0·092 for copper. It is therefore

by no means an unreasonable proposal to suggest the employment of cast copper or special copper alloy for sealing devices, obturators and possibly for parts of the cylinder head, as by its use the temperature gradient between the inside and the outside of the wall can be much reduced.

There is still scope from the viewpoints of manufacture and design for the display of greater ingenuity. The only twin rotor which has been described in the preceding pages of this treatise is the "Speedwell", attributed to the United States of America, and in the review of this engine and the discussion on page 98 it has been emphasized that the use of two separate rotors introduces an advantageous condition not found with any double-purpose inlet and exhaust valve. Positive segregation of the live gases from the exhaust, in order that contamination of the air-petrol mixture with the products of combustion cannot take place, is a principle of potential worth calling for greater attention for an important reason. The principle is one that permits the use of a liberal running clearance between the rotor and its housing, the annular space having no adverse effect on the live mixture, but merely functioning as a small local by-pass between adjacent branches of the intake manifold or the exhaust pipe, as the case may be. With a large annular clearance it is, of course, a fundamental requirement to incorporate a correctly designed sealing device in order to preserve gas tightness for compression and explosion. Such a device apparently was not envisaged in the "Speedwell" design.

The author's personal experiments cover an arrangement comprising twin cylindrical rotors, wherein each is mounted on ball bearings, one at each end of the rotor, an annular clearance being provided between rotor and housing greatly in excess of that necessary to cover differential thermal expansion and distortions. There are no fundamental disadvantages with this construction and the friction losses are determined solely by the area of the sealing surface, since all the thrust reaction due to cylinder pressures is taken on the bearings and not on the opposite side of the rotor as in most other systems of rotary valves. The sealing devices embodied in the engine for these experiments are of the piston variety, each one being provided with an ordinary compression ring and labyrinth cannelure. The illustration Fig. 116 shows in section an experimental combustion head adapted for a four-cylinder automobile engine. Since the exact area of the surface in contact with the rotor and the average pressure within the working cylinder are both known, it is a simple matter to predict the primary power-losses for this arrangement, and the computation can be carried out on a more accurate basis than is usually the case. Designs in which the friction forces are indeterminate, or where the losses cannot be calculated from the first principles of

mechanics, should be treated with suspicion. The calculation in this instance gives a loss of 0·32 h.p. for the pair of valves based on an engine speed of 4000 r.p.m. with the four cylinders developing 25 b.h.p. and with an assumed coefficient of friction $\mu = 0{\cdot}05$. The co-efficient of friction may well be very much less in actual operation, according to the mating materials employed for the rotor and sealing surface. However, accepting the coefficient of friction to be of the nature stated, it can be shown that the proportional loss of the power developed by the engine is of the order of 1·28 per cent, which compares favourably with a similar loss estab-

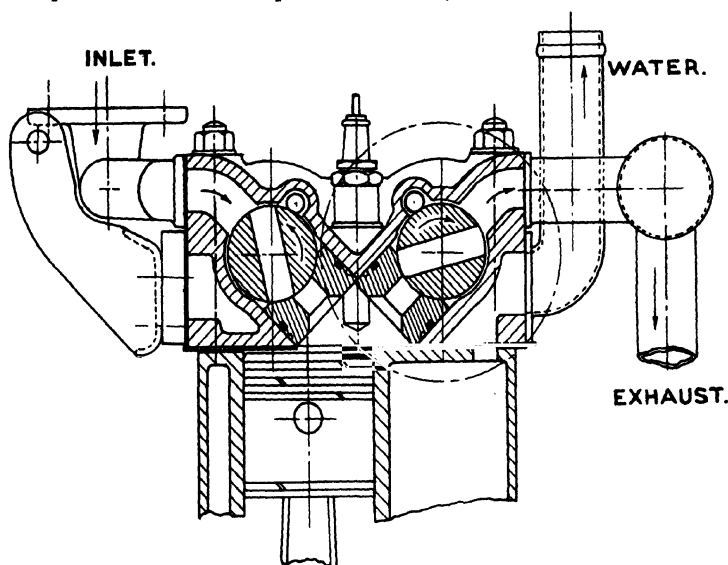



FIG. 116.
Author's experimental engine with separate cylindrical rotors for inlet and exhaust, adapted to a 900 c.c. Austin four-cylinder engine. Bore 56·77 mm. and stroke 88·9 mm.

lished by Ricardo for a poppet-valve engine. The calculation takes no account of any balancing of the pressure load on the sealing surface, which is a feasible and a logical next step in development. A practical set-up which will measure with accuracy the true h.p. absorbed by a rotary valve under actual working conditions has yet to be devised, but the problem is quite within the bounds of a solution, and experimental work in this direction will be a great help to further studies on surface friction.

It will have been gathered from previous discourse that there are a number of ways and means available to reduce the friction area of the basic piston type of obturator, as, for instance, by compensating a portion of the surface under load. Here lies an

extensive field awaiting research and practical investigation in order to determine the best proportions for optimum results.

Possibility of Valves Without Viscous Lubrication

It is of the highest importance to engineers concerned with the design of rotary valves to appreciate fully that the mechanical power-loss in driving the rotary valve is a function of sealing area, friction load and surface speed. This in some measure has been exemplified in the critical discussions on the merits or otherwise of the several designs dealt with in the foregoing chapters, but the fundamentals will bear repeating and amplifying from elementary premises.

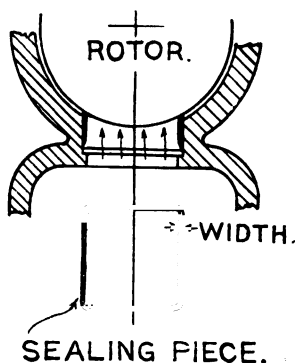



FIG. 117.
Theoretical sealing element with line contact and minimum direct loading.

Theoretically, a sealing element requires only a line contact round the port orifice, see Fig. 117. In practice a line must have some finite width, and this requirement is the first consideration which affects the area and the friction load. If the port orifice was made extremely small, then the area of the sealing surface, because of the small perimeter, would likewise be diminutive. Bearing this in mind, the magnitude of the port area is a vital factor influencing the friction. The shortest perimeter for a given area is a circle, but when the port orifice is made of circular form the rate of opening and closing speed of the valve is comparatively slow, and much of the advantage to be gained by the use of a rotary valve may be thrown away. It is therefore usual and rational to use a square or rectangular aperture. It has been shown in a previous chapter that the chordal width of port prescribes the surface speed—no matter which type of valve be chosen—and as surface speed is a factor in power-loss, then, by preference, any specified area of port is obtained

by making the chordal width small and the longitudinal dimension the greater.

Since the perimeter of the selected rectangular port is now the basis of the necessary sealing line of finite width, it is desirable to keep this width down to the smallest possible dimension and so minimize the sealing area in contact with the rotor. Practical considerations govern this width, which will vary in accordance with the strength of the material used for the sealing device; possibly two or three millimetres will be considered a suitable minimum. The smallest practical surface area for a simple sealing element is given by the length of the perimeter multiplied by its width, and the load which produces friction on the rotor is given by the sealing surface multiplied by unit pressure within the cylinder.

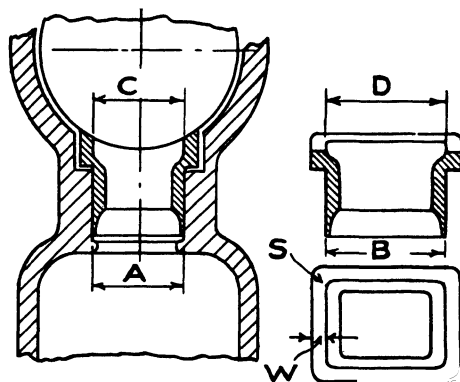



FIG. 118.
Sealing device arranged to completely balance the pressure on the friction surface.

No other friction is really necessary for the functioning of the rotary valve, if we neglect the small amount of friction from the supporting ball bearings.

Here, then, is a fundamental foundation for the almost frictionless rotary valve. How far it is possible to simulate in practice the theoretical conception depends on the ingenuity of the designer and the possible advance in manufacturing technique.

Precision methods of manufacture do undoubtedly present to the inventor greater scope for his ingenuity than in the past, and the production of unusual contours with great accuracy is now accomplished by broaching and other methods which a few years ago would have been considered quite impracticable. It is, therefore, not a far step to ask for the successful tooling of square or rectangular orifices, together with male counterparts, to a similar degree of precision as was until recently considered applicable to diametral work only.

This leads to a proposal that obturators may be fashioned with the outside contour other than circular, in which case the component may take the form of a rectangular shell of a few millimetres in thickness, and thereby replace the basic circular type of seal with a slotted orifice. Such a novel construction reduces the surface under pressure to almost negligible proportions. It should be realized at this point that however small the area of the sealing surface may be made, the quality of endurance need not be impaired, because in this type of construction the unit load per square inch is not changed, and is limited to a maximum not exceeding the peak pressure within the cylinder.

Moreover, the unit load on the running surface may, if desired, be reduced to any extent by slight variations in design. This is best demonstrated by referring to Fig. 118, where if area A × B equals area C × D, then the pressure within the cylinder will present zero load to the bearing surface of the sealing lip marked S. Obviously some small load is essential to secure the function of sealing, and the two areas may be regulated by dimensions so that the chosen unit bearing pressure is determined with great accuracy. The minimum width W of the sealing lip will be governed only by considerations of strength, no limitation being imposed by any requirements of bearing area. The problem of making the obturator gastight in the cylinder head may be solved by the feather-edge method or by progress in the arts as yet unknown.

It will now be understood that by taking full advantage of these or similar means the work-loss, and therefore the heat units to be dissipated, can be reduced to a very low value, and it may eventually be feasible to find by selection, or by development, a pair of mating materials with the necessary metallurgical properties, which will allow a rotary valve to work for the longest of periods and with good endurance characteristics, without the need for viscous lubricant, thus avoiding the complication of an oiling system with its attendant pipework, pump and regulating devices.

Sufficient has been said to indicate that the present level in design of the rotary valve falls short of what may ultimately be found possible, and it is again emphasized that many points covering design, materials and manufacture still remain to be studied. It is fervently hoped that some inspiration for further knowledge will result from reading the discussion and reflections presented in these pages.

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